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WASTE ENERGY RECOVERY STUDY

RESEARCH, ANALYSIS & DEVELOPMENT, INC.
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MAY 1978

Final Report For Period April 1977 to December 1977

AIR FORCE CIVIL ENGINEERING CENTER
(AIR FORCE ENGINEERING AND SERVICES AGENCY)

TYNDALL AIR FORCE BASE
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The report is divided into III parts. The first part outlines the techniques used for determining appropriate schemes for energy recovery. Both thermodynamic and economic considerations are presented. Part II contains rejected-energy recovery studies for facilities, which are typical at most installations. Part III is primarily concerned with special industrial recovery systems and most examples are a result of the Tinker AFB, OK, survey. Wright-Patterson AFB, OH, and Tinker AFB, OK were surveyed (was also) during this study.

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PREFACE

This report was prepared by Research, Analysis and Development, Inc. under Contract No. F08635-77-C-0126. The effort described by this report was accomplished during the period of April through December 1977 and was funded under the Investigational Engineering Program.

Mr. Robert A. Golobic was RAD, Inc's principal investigator for this study. He was assisted by Dennis A. Mrkvicka and Harold C. Schlicht of RAD, Inc. Mr. Freddie L. Beason was the AFCEC project officer for this effort. The assistance of HQ AFLC/DE, Tinker AFB/DE, and Wright-Patterson AFB/DE personnel is greatly appreciated and contributed significantly to the accomplishment of this effort.

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1. INTRODUCTION

Almost all mechanical devices in use today involve some sort of thermodynamic irreversibilities, and to the degree that a process is irreversible, it is inefficient. It may not be possible to improve the efficiency of a given device, but it may be feasible to increase the overall efficiency of the process. Rejected energy recovery is one means to this end and is part of the larger effort towards reduction in the use of high quality energy.

There are two fundamental approaches to reducing energy consumption: (1) Improving the energy efficiency of existing systems; and (2) Shifting to systems which are less energy intensive at the outset. Most techniques for improving the energy efficiency of a process tend to be capital and labor intensive, whereas alternate means of accomplishing a desired mission are energy intensive even though dollar and energy savings may result. For example, installation of storm windows reduces overall residential heat loss, increasing energy efficiency of the residence. Substituting a rapid transit system for personal transportation saves both energy and money, but is still energy intensive. In addition, dollars saved through direct energy conservation tend to be re-invested in further conservation practices, while dollars saved from substitution tend to be distributed throughout the economy in a more general or random fashion (Reference 1). In view of the evidence, one can conclude that improving the overall energy efficiency of a process is the preferred route to energy conservation. Herein lies the real significance of, and justification for, energy recovery

projects.

There are many avenues of approach to the waste energy problem. One which aids the analyst in isolating areas that exhibit potential for energy recovery is the systems approach. A systems perspective includes all aspects of energy usage and can be used to categorize energy inputs and outputs for a given building, shop, or process. For example, a boiler used for the production of steam might be considered as a system as shown in Figure 1.1.

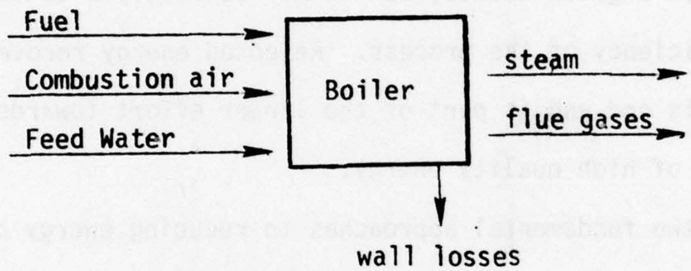

Figure 1.1. Energy transfer associated with a boiler

When inputs and outputs are systematically identified, possible sources for recovery of rejected energy become apparent. As an example, flue gases might be used for preheating combustion air and/or boiler feed water. In general, direct preheating of fuels by exhaust gases is not a safe practice, however, this might be accomplished through an intermediate heat exchange process. An added benefit of the systems approach may include opportunities for energy conservation. The example shows that energy is lost from the boiler system through structural walls. The addition of insulation could reduce wall losses, and make the overall system more energy efficient.

After possible sources for waste energy recovery have been identified, several factors must be considered before beginning energy recovery efforts.

First, the quantity of energy available should be sufficient to justify anticipated expenditures. Second, energy quality should be compatible with intended end use. Third, the recovered energy should be used in a practical and profitable way. Fourth, transmission of the energy should occur with minimal losses. Ideally, the location for the end use of recovered energy should be near the waste energy source.

Once areas for energy recovery have been isolated and the appropriate factors considered, it is beneficial to outline concepts which will yield the greatest return for a given capital outlay. Recovery schemes which exhibit the lowest cost per unit energy delivered ($UED) are preferred. In time, with increasing energy costs and changes in energy recovery technology, a recovery scheme which is presently deemed unacceptable may be practical and therefore results of the system analysis should not be discarded. The analyst should always endeavor to incorporate off-the-shelf recovery equipment. This precludes design and testing and results in the least expensive matching of source and equipment.

The first part of this report outlines the techniques used for determining appropriate schemes for energy recovery. Both thermodynamic and economic considerations are presented. These concepts are based on the fundamental principles of thermodynamics and heat transfer as found in standard texts on the subject. They are applied by surveying potential sites for rejected energy recovery at Tinker AFB, Oklahoma and Wright-Patterson AFB, Ohio. Specific examples of the methodology are given in Parts II and III of this report.

Part II contains rejected energy recovery studies for facilities

which are typical at most installations. In each case, not only are the appropriate energy fluxes considered, but standard "off-the-shelf" recovery equipment is suggested for use. These examples involve recovery from relatively low quality heat sources, since they are not associated with heavy industrial equipment. Most Air Force bases will have large quantities of low quality rejected energy; in fact, more than can be used. Every attempt to utilize these sources should be made, and a requirement for any rejected energy source in excess of 200°F can usually be found.

Part III is primarily concerned with special industrial recovery systems and most of the examples are a result of the Tinker AFB survey. The examples herein deal with high quality energy sources which, in most cases, are economically useful. Some of the sources can be directly used in the process from which they result. Others, such as the jet engine test cells, represent such a large amount of rejected energy that they can have a substantial impact on the energy utilization. Most industrial facilities within the Air Force will have the potential for rejected energy recovery equipment similar to that reported here.

PART I

2. BASIC CONSIDERATIONS IN THE UNDERSTANDING OF ENERGY TRANSFER PROCESSES

The fundamental concepts which govern the nature of energy transfer processes are the first and second laws of thermodynamics and the phenomenological relationships which govern the transfer of energy. Although this section provides a review of these concepts, one should consult a standard text on thermodynamics, such as reference 2 or 3, for more details.

Thermodynamics lends itself very well to a systems approach in identifying and discussing the problems involved with the transfer of energy. A system is something that is defined by the analyst for his particular problem at hand. The system may be either a collection of matter, or it might be a region in space. Systems range from the simplicity of a single atom, to the complexity of a nuclear power plant. Matter may flow through a system, as in the case of a jet engine. The system must be properly defined, not only to indicate the flows, but also the quality of the energy. To differentiate between various systems, the term "control mass" is used to indicate a system consisting of specified matter, and the term "control volume" is used to indicate a system of specified space. In general, when working with properties of materials, one uses the control masses. Much engineering analysis, however, involves some sort of flow process as in most applications involving rejected energy recovery. In such cases, control volumes are used. After the system is defined, everything else is automatically called the environment. The interaction between the system and

its environment is the main interest in thermodynamics.

The fundamental concept associated with thermodynamics is the first law of thermodynamics. This law states that energy is always conserved. It can be transferred from one system to another, but the sum of all the energy a system may possess and that which is transferred is always constant. Energy is usually broken into two main components, mechanical energy and internal energy. Mechanical energy consists of kinetic and potential energy and is associated with the macroscopic motion of the system or, as with a control volume, the material flowing through the system. Internal energy includes all the other kinds of energy and is usually associated with the microscopic aspects of the system. The most common form of internal energy is usually called thermal energy; that is, the part associated with the temperature of the system. Latent heat that is associated with the change in phase of a system such as the boiling of water is also internal energy. Therefore, the amount of internal energy a system possesses is not necessarily directly proportional to the temperature. Systems possess energy and transfer it from one system to another, and the energy of a system changes only when energy is transferred to or from the system. Energy is transferred through two basic modes: energy transfer as work, commonly called, simply, work, and energy transfer as heat, commonly called heat transfer. Energy transfer as work is associated with the macroscopic changes of the system, such as the volume expansion with an internal combustion engine, rotation of a shaft, or the ordered motion of electrons carrying current. These are all energy transfers as work. Energy transfer as heat is associated with microscopic motion of molecules with temperature being the driving potential. Heat transfer cannot occur without a temperature gradient. The larger the temperature gradient, the more rapid the heat

transfer rate. Thus, high temperature differences are attractive for the construction of compact heat exchangers.

Heat and work are energies in the process of being transferred. Heat and work are not stored within matter; they are "done on" or "done by" the system. Energy is what is stored. Work and heat are two ways of transferring energy across the boundaries of a system. Once energy is in the system, it is impossible to tell whether it was transferred as heat or as work. The term "heat of a substance" is thus as meaningless as "work of a substance". The first law of thermodynamics does not differentiate between the quality of the energy that has been transferred. It merely accounts for the magnitude of energy in terms of some known quantities such as a BTU, a foot-pound, a calorie, a Joule, etc.

The second law of thermodynamics describes the quality of energy in a way that the first law cannot. The quality of energy is usually measured in the ability of a system to do useful work. Work is the most precious form of energy transfer, since it can be used to accomplish virtually any task. Next comes high temperature heat sources and low temperature heat sinks; that is, those which exhibit the greatest temperature difference with the surroundings. The quality of energy transfer as heat then degenerates until its temperature difference with the surroundings is small. This is now the least precious form of energy. In most cases, when dealing with rejected energy recovery, the systems are rejecting energy as heat at relatively low quality.

Energy transfer is best used in such a manner as to preserve the quality of energy. This may seem a little obvious, but consider all the misuses of electrical energy from the thermodynamic standpoint. Electrical energy exhibits itself as work, but when used directly to space heat and/or

to heat water by resistance heaters, this most precious form has been used in the worst possible manner, that is as a low quality heat source. It has been robbed of all its availability. A better use of electrical energy for the purpose of space heating is through the use of a heat pump. In this case, the electrical energy is used to drive a compressor, and hence, work is used as work. Preserving the availability in this manner maximizes second-law efficiency. Maximizing the second-law efficiency insures that the quality of the energy source is consistent with its use (Reference 4).

In most cases, rejected energy sources are heat sources of various temperature levels. The quality of these sources must be considered in all cases. It is of little use to have a heat source of very high quantity of energy, for example several million BTUs, if these BTUs are of relatively low quality, say 100°F. All those BTUs applied to water at atmospheric pressure still do not boil the water. In considering rejected energy sources, one must therefore address not only the quantity of the energy used, but the quality of this energy. For many cases involving rejected energy recovery, the quality of the rejected energy is directly related to its temperature, and therefore provides a simple measure of energy quality.

3. AN APPROACH TO THE RECOVERY OF REJECTED ENERGY

3.1 Methodology for Conducting Rejected Energy Recovery Studies

A basic methodology for the study of rejected energy recovery has been developed from surveys done at Wright-Patterson AFB and Tinker AFB. The procedure is as follows:

1. Identify the large energy users with good recovery potential.
2. For each energy user under consideration, sketch the system and indicate the energy flows.
3. Determine the quality and quantity of rejected energy.
4. Consider the appropriate implementation of energy conservation principles.
5. Determine if the rejected energy can be coupled directly to the system.
6. Determine a location for the end use of rejected energy not coupled to the system.
7. Determine the appropriate kind of energy recovery equipment.
8. Compare costs of the various possible recovery equipment alternatives.

The identification of large energy users often leads to the identification of good sources of rejected energy. As an example, boiler flue gases are an excellent source of rejected energy. Gas-to-air heat exchange can be used for preheating combustion air. This can result in a 2% increase in combustion efficiency for each 100°F increase in air temperature (Reference 5). Gas-to-fluid heat exchangers can be employed to preheat boiler feed or

make-up water. Energy-temperature relationships are linear below the vapor dome, hence, fuel savings are also linear in this region. If the quantity and quality of the flue gases are sufficient, it may be possible to effect a heat exchange between exhaust gases and a working fluid for either conventional or organic Rankine cycle operation. Extraction of waste energy in this manner will allow production of shaft work for use as mechanical or electrical energy.

Large reciprocating air compressors are another source of rejected energy. It is common for final stage discharge temperatures to approach 300°F. A portion of the thermal energy can be extracted from the compressed air and ducted or piped to areas where heating requirements exist. Heat pipe heat exchangers are excellent for air-to-air heat recovery, while the more conventional tube and shell exchangers are best for air-to-liquid exchange.

Heating, ventilating, and air conditioning (HVAC) systems are other areas where heat recovery can be practiced in a beneficial way. The main point of concern involves pre-conditioning outside air before its entry into the structure. Heat exchange between exhaust and intake air is done to pre-cool intake air during warm months and pre-heat intake air during cooler months, reducing the load on the HVAC system. This concept can be readily applied to locations where a particular process requires large amounts of ventilation, such as plating facilities, paint spray booths, laundries, and curing ovens. Refrigeration systems are another potential source for rejected energy recovery. At present, major emphasis is placed on desuperheating compressor discharge gases before their entry into the condenser section. The amount of waste energy available varies from 3,000

to 5,000 BTUs per hour per ton, depending upon the type of condenser cooling. Energy rejected from refrigeration systems can be recovered for use in space and domestic water heating. Air conditioners that operate on a Rankine cycle have been used in a similar fashion (Reference 6).

Once a system has been selected as a potential site for energy recovery, sketch the system and indicate the flows of energy. The energy flows should be identified by energy transfer as heat or energy transfer as work. In almost every case, energy transfer for rejected energy purposes will be energy transfer as heat. This sketch often provides the groundwork for immediate implementation of rejected energy recovery equipment, and therefore is one of the most important steps in the analysis.

Once the flows of energy have been indicated, it is necessary to quantify each of the rejected energy sources. For the base studies, this has proven to be an arduous task. Rejected energy data have not been available because, when most of the facilities were built, the cost of energy was low, and no metering devices were installed. Therefore, the quality and quantity of the rejected energy must be determined with best data estimates. In most cases, the determination of quality merely involves the determination of the temperature of the rejected energy source. When mass is rejected, it also involves the determination of pressure. In addition, any corrosive conditions that exist within the rejected energy source should be noted because this will influence the kind of energy recovery equipment that may be used. In addition to determining the magnitude and quality of the energy, the period over which a particular facility generates rejected energy also needs to be determined.

The first step in the use of rejected energy is to determine if energy can be conserved rather than used as a rejected energy source. Can the overall

energy consumption of the particular system or device be lowered through better conservation practices? The implementation of good conservation is usually less expensive and pays back more rapidly than does the implementation of rejected energy recovery systems. For example, it makes little sense to use flue gases in a gas furnace to preheat combustion air -- which will result in perhaps a 4% savings in energy -- when the ductwork for that particular distribution system is leaky and not insulated. Such a distribution system could have up to 40% energy losses. In this case, it is best to conserve energy by insulating the ductwork or repairing the leaks and then preheat combustion air.

After appropriate conservation principles have been applied, the system should be examined to determine whether rejected energy can be directly used by the system; as is the case for exhaust gases on a furnace preheating combustion air. When this is possible, rejected energy is fully used, because it will be utilized when the system is in operation. Rejected energy systems such as these usually demonstrate rapid payback periods.

When rejected energy cannot be coupled directly to the system, an appropriate location for its use must be found. Three principal factors are to be considered. The first of these is the amount of use a location will have for the source. The second being the physical distance that the energy must be transported for it to be utilized. The third factor is the maintenance required for the energy recovery system. Energy recovery studies have indicated that these three factors are critical; if a source is used only slightly, if the distance of transmission is quite long, or if large maintenance costs are required, the location for end use will usually not be economically feasible.

After the possible locations for end use of the rejected energy have been determined, the type of equipment should be considered. It is best to consider devices which can be purchased "off-the-shelf", because these are usually initially cheaper than the "custom made" devices. Several types of heat recovery devices are available, each having particular advantages and disadvantages. Consequently, recovery conditions and parameters may simplify the overall selection by elimination of choices. A list of recovery hardware includes enthalpy wheels, rotary heat exchangers, heat pipes, run-around coils, and various types of shell and tube heat exchangers. Enthalpy wheels can recover both latent and sensible heat, but an external energy source is required to impart rotary motion. Cross-contamination may also occur between the supply and exhaust streams. In addition, the two streams must be physically close. Rotary heat wheels are essentially enthalpy wheels, but without the dessicant coating, and therefore can recover only sensible heat. The disadvantages associated with rotary heat wheels are similar to those associated with enthalpy wheels. Heat pipes are entirely static devices and require no external energy input. They can recover both sensible heat and latent heat, but require the exhaust and supply streams to be in close proximity. Run-around coils consist of two heat exchangers which can be separated by a relatively large physical distance. The exchangers are interconnected by piping filled with a working fluid which acts as a heat transfer medium. The fluid is circulated by an electrically driven pump requiring an external power supply, although the pump is usually quite small. Shell and tube heat exchangers are the classical types of devices used for energy recovery and are usually used for gas-to-

liquid heat exchange.

Other devices worth considering from the energy recovery viewpoint are organic Rankine cycle units and waste heat boilers. Organic Rankine cycle units can be used to produce shaft work or generate electricity and can use waste heat streams as a thermal driving source. Because organic working fluids change phase at much lower temperatures than water and possess higher densities, these units can be quite successful for producing shaft work in waste heat recovery systems. Waste heat boilers are simply devices that capture waste heat and use it to provide heating or cooling, or to act as a heat source for the Rankine cycle process.

After the locations for potential heat recovery have been determined and the appropriate types of energy recovery units have been selected, the costs associated with the recovery equipment and the potential energy savings from using this equipment can be determined. This is the key step in the analysis, because it will indicate the payback period of the device. The typical economic analysis for these systems is the present sum analysis. Basic economic considerations for this type of analysis are described in Section 4. If the payback period is relatively short, say for instance, three to four years, then one should consider implementing the device. The final consideration involved is a measure of what is commonly known in engineering economics as an irreducible. For example, if a rejected energy recovery system has a payback period of three to four years, but the facility from which the rejected energy is coming is to be replaced in the next two years, it makes no sense to install the rejected energy recovery equipment.

Data collection and/or metering devices aid in the proper determination of rejected energy recovery equipment. The installation of such equipment

will assist in isolating areas that are major energy consumers. The added benefit associated with metering includes the ability to document energy reductions that occur as a result of energy conservation measures.

3.2 Waste Energy Recovery Survey Questionnaire

One of the most difficult aspects of applying the methodology is usually the identification and quantification of waste energy sources. As a result of the base surveys, the questionnaire shown at the end of this section has been developed, which should assist in this process. It is divided into four sections, and in most cases, not all sections will apply. The form has been designed to suffice for a single building. In the case of a building such as Building 3001 at Tinker AFB, the multitude of industrial processes located within the building require more than a single form. For cases such as this, it is best to have one form per shop. The Building Load Information Surveys (BLIS) and first-hand knowledge of various buildings on a base should provide the initial data before surveys are conducted. With this input, first possible locations for waste heat recovery can be determined.

The first section of the form is concerned with forced air and ventilation systems. The first two blocks require information about the ventilation requirements. Those facilities which require the largest number of air changes per hour or have the inlet/outlet centrally-located and do not already possess heat recovery units (Block I.4.) should be examined first. Processes which require separate exhaust ventilation often rely upon infiltration for intake, those which require the largest flow rate should also be examined first. As noted in the previous section, uninsulated ductwork passing through spaces which do not require heating or cooling can cause excessive losses. Block I.3. therefore provides for the delineation of such conditions. Block I.5 refers to the support of a process

that requires chilling or refrigeration other than air conditioning.

Section II relates to any device located within the building or area of interest that exhibits energy transfer as heat to the surroundings and/or exhausts hot gases, for example, forced air gas furnaces, treating ovens, boilers, industrial furnaces, etc. Since utilization has been found to be an important parameter with regard to the economics of heat recovery, Block II.1.d. provides for the rating of utilization. Any convenient time scale is appropriate, such as hrs/week, mos/year, etc. Blocks II.2., II.3., and II.4. provide for noting the presence or absence of insulated exhaust stacks, combustion air preheating and any existing heat recovery units which have all been found to be successful fundamental conservation and recovery schemes.

Section III is reserved for the discharge of hot liquids such as those from cooling systems, condensate and compressed air lines. Such lines may prove useful for recuperative heat exchange, space heating, and/or potable water heating. Existing insulation is recorded in Block III.2. The relative merits of line insulation is discussed in Section II. Block III.3. permits the noting of any existing recovery systems.

Section IV provides for the delineation of major electric devices. While lighting and air conditioning units may be the most common, it also provides for devices such as air compressors and electric furnaces. Examination of the lighting system as required in Block IV.1. may expose some abuses such as excess lighting in areas originally designed for one purpose and now being used for another. Major electric devices (Block IV.2.) which dissipate energy for a useful purpose are often good low quality heat sources.

Section V is provided for additional comments which may include devices not covered in the previous four sections, ideas for possible recovery schemes, and requirements germane to the building or area which may preclude the use of heat recovery equipment.

This sheet is a product of the surveys and should assist a base civil engineer and/or base maintenance personnel in the identification of sources for heat recovery. Information gathered as a result of this sheet report should provide the initial input for a system approach to the recovery of rejected energy.

BUILDING	SHOP	NAME	OFFICE SYMBOL	PHONE NUMBER

SECTION I - FORCED AIR AND VENTILATION SYSTEMS

1. Amount of outside air required (CFM)	Summer
	Winter

2. Processes which require separate ventilation

a. process	b. exhaust temperature	c. flow rate (CFM)

3. HVAC Ductwork insulated	yes	no	some (specify amt)
4. Heat recovery units in use	type	process	none
5. Refrigeration/chiller requirement	purpose	rating	none

SECTION II - HEAT SOURCES (GASES)

1. Heat sources in the area

a. type	b. fuel	c. temperature	d. utilization (time)

2. Exhaust system insulated	yes	no	N/A
3. Combustion air preheating	unit	none	N/A
4. Heat recovery units in use	type	process	none

SECTION III - HOT FLUID EFFLUENTS (LIQUIDS OR COMPRESSED GASES)

1. Hot effluents (> 100°F)

a. fluid	b. flow rate	c. temperature	d. line size

2. Lines insulated	yes	no	some

3. Heat recovery units in use	type	process	none

SECTION IV - ELECTRICAL DEVICES

1. Electric lighting supplemented by natural lighting	type	none

2. Major electric devices

a. Name	b. Purpose	c. kw/HP rating

SECTION V - ADDITIONAL COMMENTS

4. ECONOMIC EVALUATION OF WASTE ENERGY RECOVERY EQUIPMENT

4.1 Present Worth Economics

The most popular technique for evaluating energy saving equipment is through the use of present worth economics. In this case, all costs and savings are represented as if they were to occur on the day of installation. The appropriate use of rates-of-return of compound interest factors are required to account for the time value of money. Economics for rejected energy recovery equipment involve two basic costs, the initial capital outlay, C, and the annual maintenance costs, M_n, where n refers to the number of years after installation. The savings principally involve a decrease in the cost of some utility; this is represented by the symbol G_o, that is the annual utility savings associated with the year prior to installation. Other costs may be involved such as other utility savings or reduced equipment operation costs, and should be properly accounted for. In general, however, it has been found that these costs are relatively small and therefore are not included here. The basic equation for present-worth break-even analysis is

$$P = C + \sum_{j=1}^{n} \frac{M_n(1 + i_i)^j}{(1 + i^*)^j} - \sum_{j=1}^{n} \frac{G_o(1 + i_u)^j}{(1 + i^*)^j} \leq 0 \qquad (4\text{-}1)$$

where i_i is the inflation rate of the maintenance costs and i^* the prescribed rate-of-return, and i_u the escalation rate for the utility of interest. The annual costs are computed for an entire year and are

considered to be a year-end cost. Typically, the unknown in the equation is n, the break-even year. This equation is implicitly solved for the value of n which makes P less than or equal to zero. Assuming a rate-of-return of 7%, and because this is very nearly the inflation rate, the second term in the equation is simplified. In addition, the annual maintenance costs are usually best approximated by a constant. With the above assumptions, equation (4-1) becomes

$$P = C + n\,M - \sum_{j=1}^{n} G_0\left(\frac{1 + i_u}{1 + i^*}\right)^j \leq 0 \tag{4-2}$$

The utility escalation rates are typically greater than the rate-of-return so a similar approximation cannot be made for the last term in equation (4-2). Table 4.1 gives utility escalation rates used throughout the report.

	Period	i_u	Period	i_u
Natural Gas	1977 to 1980	15%	1981 to ∞	8%
Oil*	1978 to 1980	16%	1981 to ∞	8%
Coal	1977 to 1980	10%	1981 to ∞	5%
Electricity	1977 to 1980	16%	1981 to ∞	7%

*1977 rate 12%

Table 4.1. Utility escalation rates.

The summation of equation (4-2) can be replaced with a single term if the following substitution is made

$$\frac{1 + i_u}{1 + i^*} = 1 + I \tag{4-3}$$

where

$$I = \frac{1 + i_u}{1 + i^*} - 1 \tag{4-4}$$

The summation is then

$$\sum_{j=1}^{n} (1 + I)^j = \frac{(1 + I)[(1 + I)^n - 1]}{I} \tag{4-5}$$

for first value of j not equal to 1

$$\sum_{j=m}^{n} (1 + I)^j = \frac{(1 + I)[(1 + I)^n - (1 + I)^{m-1}]}{I} \tag{4-6}$$

when $i_u = i^*$ $\quad I = 0$, and

equation (4-5) becomes

$$\sum_{j=1}^{n} (1+I)^j = \sum_{j=1}^{n} 1 = n \tag{4-7}$$

equation (4-6) becomes

$$\sum_{j=m}^{n} (1 + I)^{j} = \sum_{j=m}^{n} 1 = n - m - 1 \tag{4-8}$$

Table 4.2 gives the values for I for an i^* of 7% and values of i_u from Table 4.1.

	Period	I	Period	I
Natural Gas	1977 to 1980	7.5%	1981 to ∞	.93%
Oil*	1978 to 1980	8.4%	1981 to ∞	.93%
Coal	1977 to 1980	2.8%	1981 to ∞	-1.9%
Electricity	1977 to 1980	8.4%	1981 to ∞	0%

*1977, I = 4.7%

Table 4.2. Values of I for the various utility escalation rates.

Most economic evaluations were conducted assuming an installation of January 1, 1978, therefore two rates were involved. For $n \leq 3$, equation (4-2) becomes:

$$P = C + n\,M - \frac{G_0(1 + I_1)[(1 + I_1)^{n} - 1]}{I_1} \leq 0; \; n \leq 3 \tag{4-8}$$

For $n > 3$

for natural gas, oil and coal

$$P = C + n\,M - G_o\left\{\frac{(1 + I_1)[(1 + I_1)^3 - 1]}{I_1} + \frac{(1 + I_2)[(1 + I_2)^n - (1 + I_2)^3]}{I_2}\right\} \leq 0 \tag{4-9}$$

for electricity

$$P = C + M - G_o\left\{\frac{(1 + I_1)[(1 + I_1)^n - 1]}{I_1} + (n - 3)\right\} \leq 0 \tag{4-10}$$

These are the governing equations for the economic evaluation of the energy recovery equipment.

4.2 The Effect of Uncertainties on the Determination of Break-Even Time for Waste Energy Recovery Equipment

The computation of the break-even economics depends to some measure upon the ability to predict utility escalation rates, projected maintenance costs, and to some extent the projected utility savings and the initial capital investment. Although statistical methods exist (Reference 7), the effects of these uncertainties can be clearly described through the logarithmic derivative of the present-worth break-even equation.

The significance of logarithmic differentiation is shown by the following example. Consider a dependent variable, z, which is functionally related to an independent variable, x, as given below

$$z = f(x) \tag{4-11}$$

Logarithmically,

$$\ln z = \ln (f(x)) \tag{4-12}$$

The derivative of equation (4-12) is

$$\frac{dz}{z} = \frac{f'(x)}{f(x)} dx \tag{4-13}$$

Multiplying the righthand side by x/x, the result is

$$\frac{dz}{z} = \left[x \frac{f'(x)}{f(x)} \quad \frac{dx}{x}\right] \tag{4-14}$$

dx/x and dz/z are non-dimensional differences or uncertainties that are expressed in terms of percents or percentages. The term in the brackets

is called the sensitivity coefficient because it functionally relates dx/x to dz/z. Suppose x has a value of 10 and this value is known to be within ± 1. The uncertainty in x is therefore 10%. Since z is a function x, this uncertainty is quantitatively projected to z through the sensitivity coefficient. If the value of the sensitivity coefficient were to be one, then a 10% uncertainty in x would translate to a 10% uncertainty in z. If the coefficient were 10 or .1, then the uncertainty in z would be 100% or 1%, respectively. The sensitivity coefficient therefore indicates the effect of a particular independent variable on the dependent variable.

The result of equation (4-14) can be extended to two or more variables.

$$z = f\,(x, y) \tag{4-15}$$

The logarithmic derivative is

$$\frac{dz}{z} = \left(\frac{x}{f(x, y)}\right)\left(\frac{\partial f}{\partial x}\right)_y \frac{dx}{x} + \frac{y}{f(x, y)}\left(\frac{\partial f}{\partial y}\right)_x \frac{dy}{y} \tag{4-16}$$

Extending to n number of variables, the sensitivity coefficient, S_i, takes the form

$$S_i = \frac{x_i}{f(\mathbf{X})}\left(\frac{\partial f}{\partial x_i}\right), \text{ where } \mathbf{X} = x_i, x_2, \ldots, x_n \tag{4-17}$$

The equation of interest in the economic analysis is the present worth break-even equation (c.f. (4-8))

$$P = C + n\,M - \frac{G_0\,(1+I)\,[(1+I)^n - 1]}{I} = 0 \tag{4-18}$$

In this case, there are five independent variables, C, M, G_o, i_u and i^*.

The dependent variable is n, and although equation (4-11) cannot be solved explicitly for n, it is possible to determine the derivatives of interest. The general logarithmic derivative which describe changes in n is

$$\frac{dn}{n} = \frac{C}{n}\left(\frac{\partial n}{\partial C}\right)\frac{dC}{C} \pm \frac{M}{n}\left(\frac{\partial n}{\partial M}\right)\frac{dM}{M} + \frac{i^*}{n}\left(\frac{\partial n}{\partial i^*}\right)\frac{di^*}{i^*}$$

$$+ \frac{i_u}{n}\left(\frac{\partial n}{\partial i_u}\right)\frac{di_u}{i_u} + \frac{G_o}{n}\left(\frac{\partial n}{\partial G_o}\right)\frac{dG_o}{G_o} \tag{4-19}$$

The derivatives required by equation (4-19) have been derived in Appendix A. The results of which are presented here.

$$\frac{dn}{n} = [n(F_1 - \frac{M}{G_o})]^{-1}[\frac{C}{G_c}\frac{dC}{C} + n\frac{M}{G_o}\frac{dM}{M} + \frac{i^* F_2(1+I)}{(1+i_c)}\frac{di^*}{i^*}$$

$$- \frac{i_u F_2}{(1+i_c)}\frac{di_u}{i_u} - F_3\frac{dG_o}{G_o}] \tag{4-20}$$

where

$$F_1 = \frac{(1+I)^{n+1}}{I}\ln(1+I) \tag{4-21}$$

$$F_2 = \frac{(nI-1)(1+I)^n + 1}{I^2} \tag{4-22}$$

$$F_3 = \frac{(1+I)[(1+I)^n - 1]}{I} \tag{4-23}$$

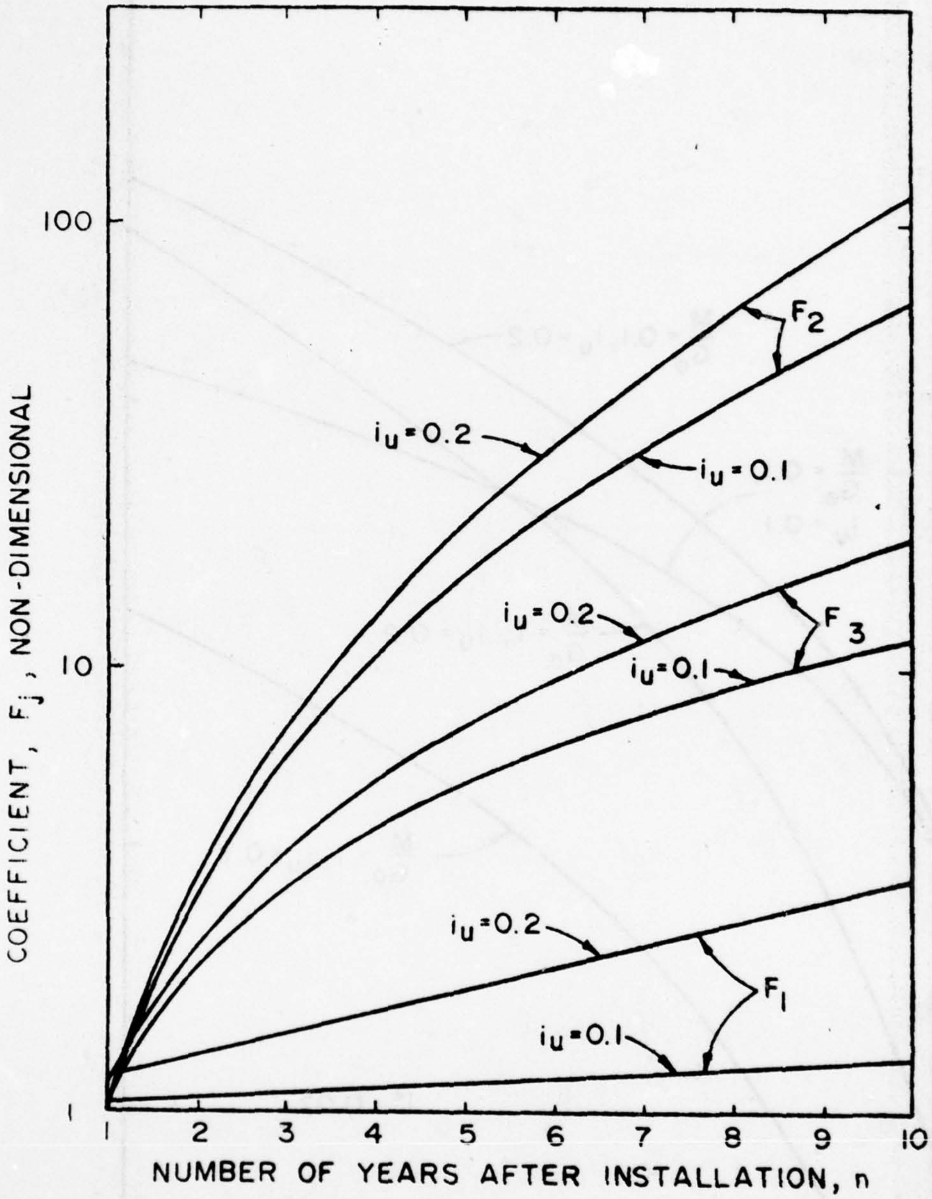

Figure 4.1. Non-Dimensional functionals for a rate of return of 7%.

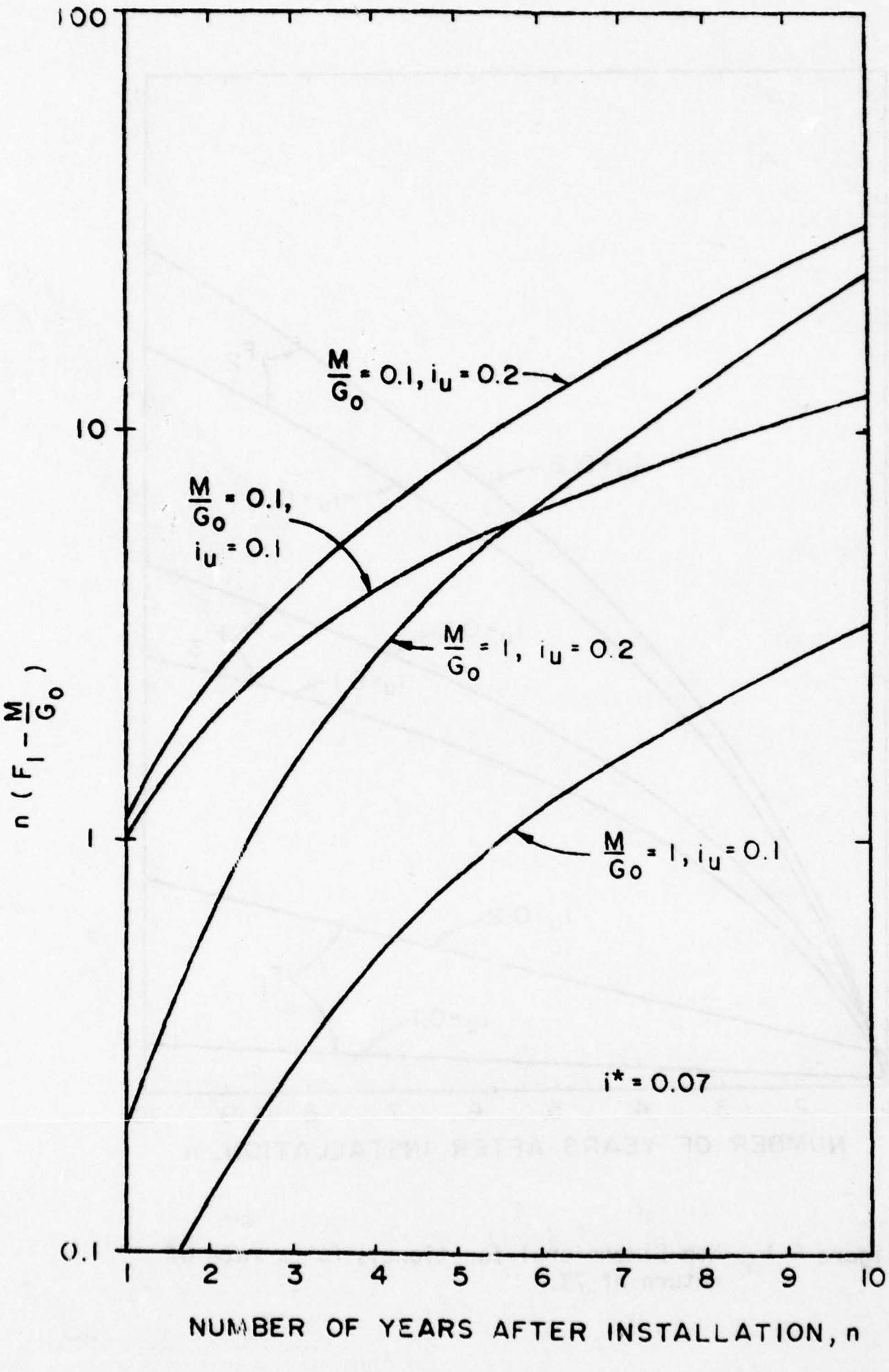

Figure 4.2. $n(F_1 - M/G_o)$ for various values of M/G_o and i_u as a function of number of years after installation.

The factors F_1, F_2 and F_3 are shown in Figure 4.1 as a function of the number of years, n. Note that the factor $n[F_1 - \frac{M}{G_0}]$ divides each one of the coefficients. Representative values for this factor are shown in Figure 4.2.

Note that only the capital cost coefficient and the maintenance cost coefficient of equation (4-20) contain terms which directly relate to the dollars expended for the particular device. If the capital cost of a particular energy system is uncertain by 10%, that is, $dC/C = 0.1$, and if the coefficient $C/(G_0 n(F_1 - M/G_0))$ is equal to one, then this uncertainty contributes a 10% uncertainty to the break-even time n. If the coefficient were equal to 10, then a 10% uncertainty in the capital cost leads to a 100% uncertainty in the break-even time. Similarly, if the coefficient were equal to 0.1, then a 10% uncertainty in the capital cost leads to a 1% uncertainty in the break-even time. The sign on the coefficient indicates the direction of the change. For instance, an increase in the energy cost savings, G_0, leads to a decrease in the break-even time, because the sign on F_3 is always negative.

The term $(F_1 - M/G_0)$ which appears in the denominator of each term on the righthand side of Equation (4-20) can have a significant effect on the uncertainty if M/G_0 is on the order of F_1. Since F_1 always has a value of one or greater, M/G_0 must be kept small in order to keep uncertainty low. The maintenance costs of a particular unit should therefore be much less than the projected cost savings of energy at today's prices. Intuitively, one may have guessed this result, because systems which require maintenance approximating projected savings do not appear to be a sound investment. Having a small value of M/G_0 minimizes the effect of the

$(F_1 - M/G_o)$ term, and insures that the coefficient of the dM/M term is small. One may conclude, then, if maintenance costs are small compared to utility cost savings, uncertainties in maintenance costs lead to relatively little uncertainty in the break-even time. A similar conclusion may be drawn with regard to capital investment which is the first term in Equation (4-20), the difference being that it is usually possible to have accurate estimates of capital costs. When the C/G_o ratio is large, every attempt must be made to accurately determine the initial capital costs so that break-even time can be accurately estimated. Energy recovery systems which produce work will, in general, have a relatively large capital investment compared to utility cost savings.

The relative values of the remaining three coefficients are shown in Figure 4.3. The lowest pair of curves in Figure 4.3 are for the coefficient of the di^*/i^* term which is the inflation or cost-of-money term. It indicates that a change in the inflation rate has a relatively small effect on the uncertainty in the break-even time. For example, a 50% uncertainty in the inflation rate would lead to about a 10% uncertainty (For $i_u = .2$) if the break-even time were five years. Thus, if the inflation rate were 10.5% instead of 7% (50% higher), then the break-even time would be 5.5 years instead of five years. It appears, therefore, that inflation rate uncertainties are not a cause for concern in projected break-even times.

The utility savings term coefficient, $d\,G_o/G_o$ is on the order of one and therefore a 20% uncertainty in this value will lead to about 20% uncertainty in the break-even time. For example, if the utility savings were actually 20% lower than predicted, a five year break-even time would increase

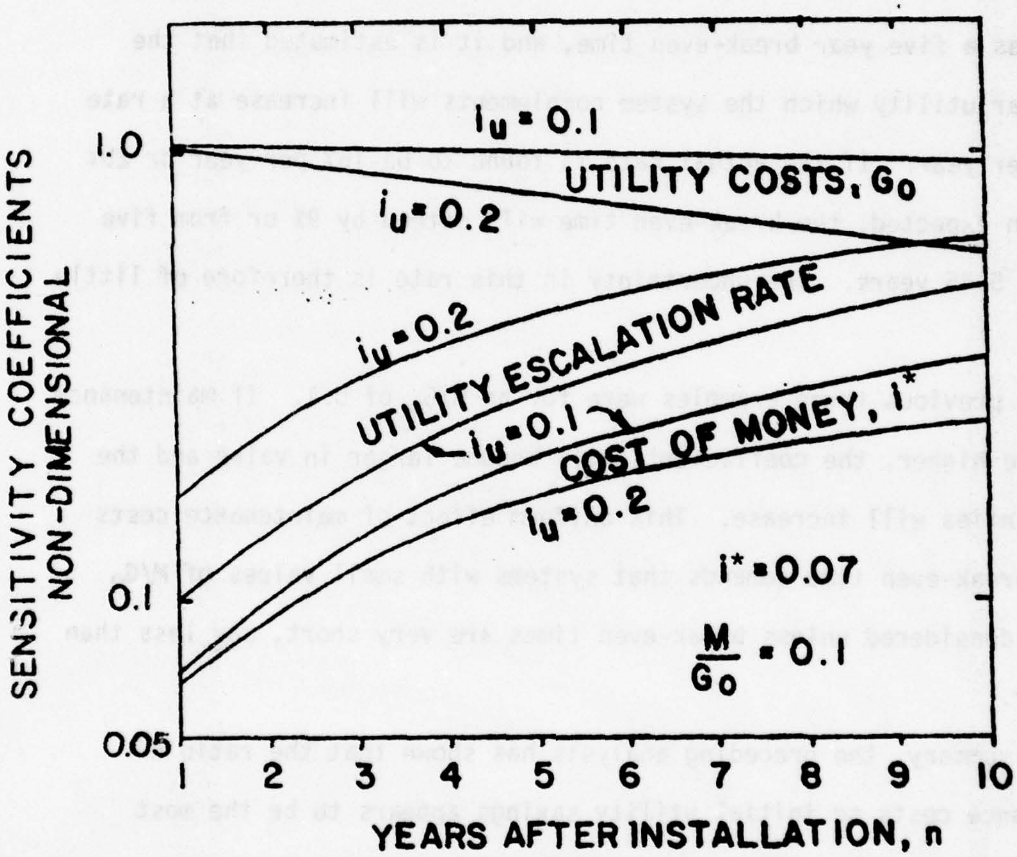

Figure 4.3. Sensitivity coefficients as a function of years after installation.

to six years. It is expected that the utility savings estimate would be within 10% of projected value. Five-year break-even estimates therefore would have a $\pm$ .5 year uncertainty.

The uncertainty involved in knowledge of the increase in utility costs appears to yield a minor sensitivity. Suppose an energy recovery system has a five year break-even time, and it is estimated that the particular utility which the system complements will increase at a rate of 20% per year. If the actual rate is found to be 16% per year or 20% less than expected, the break-even time will extend by 9% or from five years to 5.45 years. The uncertainty in this rate is therefore of little concern.

The previous three examples were for an M/G_0 of 0.1. If maintenance costs are higher, the coefficients will become larger in value and the uncertainties will increase. This uniform effect of maintenance costs on the break-even time demands that systems with small values of M/G_0 only be considered unless break-even times are very short, say less than one year.

In summary, the preceding analysis has shown that the ratio of maintenance costs to initial utility savings appears to be the most critical parameter in the uncertainty in estimating break-even costs. If this ratio is small, then the ratio of capital outlay to initial utility savings appears to be the most critical. The analysis indicates that for systems requiring a relatively large capital outlay, the cost estimates should be known accurately in order for accurate estimates in break-even time to be made. Since uncertainties in utility savings project themselves to comparable uncertainties in break-even times,

systems with large uncertainties in utility savings should be reviewed subjectively so that one can insure that the derived benefits will outweigh any criticism which may arise in future years. It must be understood that uncertainties in cost-of-money and utility escalation rate have small effects on the uncertainty in the break-even time.

PART II

5. RAISING THE EFFICIENCY OF GAS FURNACES

In this section the techniques for improving the performance of ordinary forced-air gas furnaces are discussed. Heat recovery in this case is simplified, because the furnace itself is directly utilizing the recovered energy. The source and the demand are therefore directly coupled. Furnace usage varies throughout the continental United States, and the average is likely no more than six months. Break-even times for heating equipment in general can therefore be expected to be longer than similar equipment used year round. The absolute savings for a particular furnace may be small. For example, increasing the efficiency by 10% may save only from $20 to $50 per year, but considering the number of furnaces in use at Air Force installations, the resulting savings will be significant.

Recent studies and experiments carried out by the Oak Ridge National Laboratory (Reference 8) have illuminated possible areas for gas furnace system energy savings. According to the report, the greatest controllable loss from a forced hot-air heating system is from uninsulated or leaking ducts. The problem can be remedied by examination and repair of ducts and insulation as necessary. In some residences where ducting passes through unheated space, losses as high as 40% were reported.

The second greatest controllable loss (up to 10%) results from the cyclic operation of the furnace. The report suggests that this loss can be virtually eliminated by setting the fan switch to turn the fan off at

$5^{O}F$ above the room thermostat setting and to turn it on as closely above the setting as the switch construction permits. This involves a simple modification and comes at the expense of some comfort.

The report also recommends turning off the pilot flame (10% loss) during nonheating seasons and performing routine cleaning and adjustments as suggested by the equipment manufacturer. In addition, when the main furnace control valve needs replacing, replace it with the valve pilot combination that can automatically turn the pilot light on when needed. Such combinations cost about $10 more than a regular system, but easily pay for themselves in one heating season.

Preheating combustion air for gas and oil fired furnaces and/or boilers can reduce stack losses from 10% to 25%. For small applications (exhaust stack diameter: $6" \leq x \leq 8"$), a heat pipe heat exchanger is suitable. For larger units, conventional tube type heat exchangers are probably a better choice. In any case, furnace efficiency can be improved by 2% for each $100^{O}F$ rise in combustion air temperature with a $200^{O}F$ rise typical in most applications. Alternately, heat can be drawn from the system for cold weather heating by placing forced draft heat pipe heat exchangers in the exhaust ducting. These units can be thermostatically controlled and demonstrate per unit heat recovery potentials of 5000 to 15000 BTU/hr depending upon furnace capacity, size of heat exchanger (i.e., number of heat pipes in unit), and the exhaust temperature. Each unit can supply warmed air at temperatures between $140^{O}F$ - $200^{O}F$ with air flow rates of 80 to 100 cubic feet/min. An example of the type of heat pipe heat exchanger that would find application in this instance is the Air-O-Space heater, distributed by Isothermics, Inc., of Augusta, New Jersey. It can be installed in ducting diameters between

six and eight inches with optimal temperature ranges of 450°F to 800°F.

It appears, therefore, that in the case of furnaces, conservation should be practiced first. Turning off pilot lights in the summer and reducing the cyclic operation of the furnace by permitting broader temperature range can be accomplished at virtually no cost and will provide an immediate 10% to 20% savings in the gas utilized. Insulating ductwork in unheated spaces and repairing leaks in the ducting will then provide additional savings. Only after these conservation measures are adopted should the recovery of stack gas energy be accomplished. Recovery equipment costs from $100 to $500 and the investment will usually be recovered in less than five years.

6. FREE COOLING FOR LARGE TRANE CHILLER UNITS

"Free cooling" refers to a TRANE air conditioning chiller operating mode wherein the motor driven compressor can be electrically de-energized. The system can then be operated as a basic heat exchanging unit and produce up to 45% of its rated cooling capacity. Figure 6.1 illustrates the normal operating flowpath for the unit, with the electrically driven compressor providing work input to the system. Figure 6.2 demonstrates the free-cooling mode, flow being maintained by natural convection, with the compressor de-energized.

The free cooling mode can be initiated whenever cooling water temperature is less than the chilled water replacement. Refrigerant is boiled in the evaporator, absorbing heat from the chilled water. The vapor flows to the condensing section where it is condensed, and returns to the evaporator section by gravity flow. The performance of the chiller unit is a function of cooling water temperature and flow rate, and the amount of surface area available for heat transfer. As stated, the chiller can operate up to 45% of its rated capacity in the free cooling mode for units of 200 to 1360 tons. The KW requirements for these units vary from 150 to 1000 KW. Clearly, electrical savings can be significant. At a 1977 Air Force electric rate of roughly 2¢ per kilowatt hour, the savings on operating a 150 KW motor for 8 hours per day, 120 days per year, is $2,880 while a savings of $19,200 is possible for a 1000 KW unit.

It is recommended that the free cooling option be considered only

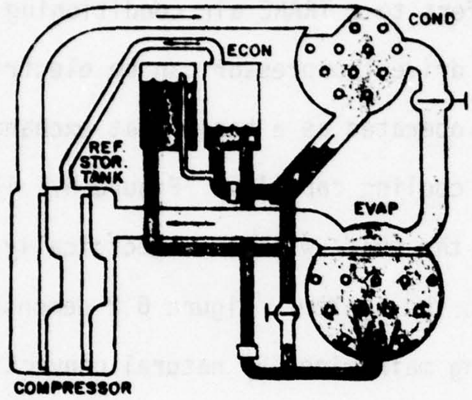

Figure 6.1 Normal chiller operation

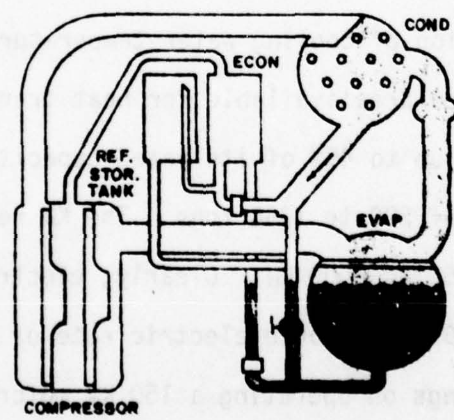

Figure 6.2 Free cooling mode

in geographical areas which average greater than 4,000 degree days. In addition, the system is recommended where precisely controlled humidity is not critical, or where outside air cannot be used satisfactorily.

7. ORGANIC RANKINE CYCLE WASTE HEAT UTILIZATION

An organic Rankine cycle energy conversion system is simply a Rankine cycle that uses freon or similar organic fluid as a working fluid. Since the required boiler temperatures are much lower than for Rankine-cycle steam units, "waste" heat can be used as the source when the production of work is determined to be economically acceptable. The basic process is illustrated in Figure 7.1.

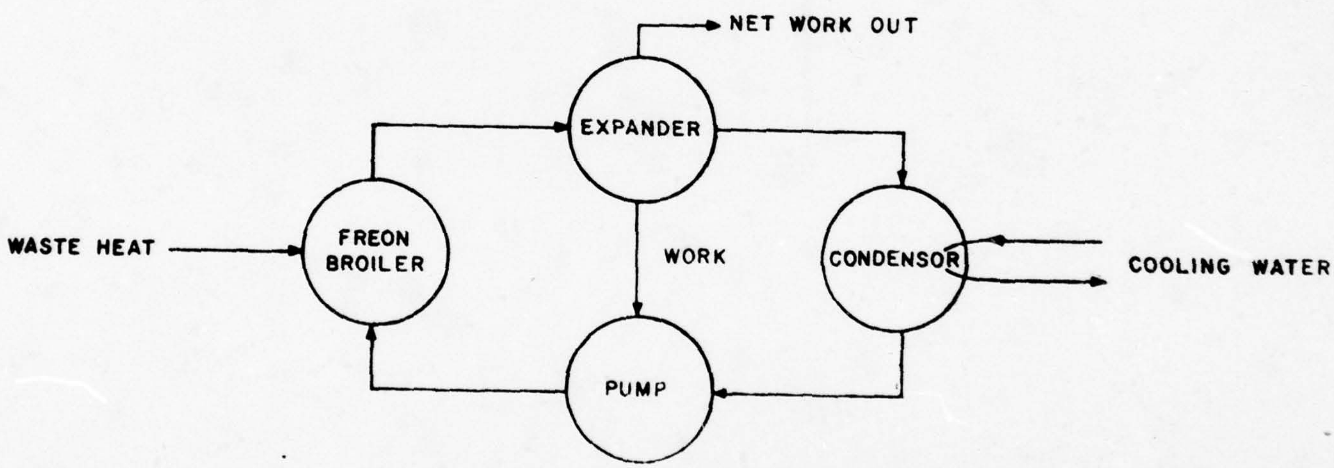

Figure 7.1 Basic Organic Rankine Cycle

Parameters associated with the waste heat source (mass flow rate, temperature, etc.) are the primary limiting constraints to power output, while condensing fluid temperature is the second major constraint. Energy extracted from the waste heat stream in the freon boiler causes vaporization of the working fluid. The vapor then passes through the expander where shaft work is produced, while the condenser section condenses the working fluid and the pump returns it to the boiler where the process continues.

Large units (600 KW) such as those manufactured by Sunstrand Energy Systems of Rockford, Illinois, are still in the developmental stage and the present unit cost is about \$310,000 with installation requiring an additional \$100,000 to \$150,000. The waste heat source temperature for optimal operation is in excess of 600°F with a mass flow rate of about 80,000 pounds mass per hour. At this point in time, units with this capacity would not seem justifiable, because of capital commitment required by Air Force mission and needs.

Smaller units (10-15 KW) may be justifiable. One manufacturer of these units is Sun Power Systems Incorporated of Miami, Florida. These units can use a liquid or gaseous waste heat source with temperatures greater than or equal to 150°F and can produce 10 KW with a boiler/condenser temperature differential of 100°F. The maximum power capability is 15 KW and the minimum is 3 KW at a ΔT of 50°F.

An organic Rankine cycle (ORC) has several advantages over conventional steam systems including low flow rate to power production potential ratio allowing smaller expansion engines, piping, and condensers.

In addition, erosion of turbine blading during the freon expansion process is minimized due to freon's ability to remain saturated during the enthalpy reduction taking place in the turbine. Freon velocity is characteristically lower than steam for a given work output and it can develop greater pressures for a given temperature than steam, especially for the lower temperature values typically encountered in waste heat applications. Based upon the Carnot limit, the process becomes more efficient with increases in temperature of the waste heat source and decreases in the temperature of the cooling source. However, an organic heat engine is more efficient than common steam systems where heat source temperatures are limited.

A typical 10 KW ORC unit has the following dimensions: length 96", width 60", height 48", weight 2627 pounds. The shaft work produced by the unit can be used directly or applied as a prime mover for an electrical generator. Cooling water can be supplied by either open or closed circuit cooling systems with a typical condenser ΔT of $18^{\circ}F$ at a 50 gpm flow rate.

These units could find wide application throughout the Air Force, particularly at central heating plants. For example, hot stack gases could be used as the heat source for operating the unit, while the shaft work produced could be used directly by mechanically coupling the engine output to a feed pump, condensate pump or circulating pump in the case of steam plants, or circulating pumps where high temperature hot water is produced. These suggestions are representative only and are not intended to include all possible uses for an organic Rankine cycle (ORC) power system.

The cost of a 10 KW ORC unit is approximately $11,500. The unit will require a heating and cooling supply. If circulating water is already available, the cost of piping water to the unit should be relatively small. If no cooling supply presently exists, a cooling water circulating system would have to be installed. For a unit of this capacity, any closed circuit evaporative air cooler would suffice with approximate costs $2,750. The waste heat source (exhaust stack effluent) would require a heat exchanger placed in the exhaust stream and the exchange media could be air or fluid. Pricing of the heat exchanger will require further investigation if interest warrants, but a cost of $2,000 would be a reasonable estimate. The unit is virtually maintenance free for 30,000 hours of operation and maintenance costs will therefore be considered negligible.

An economic analysis of a hypothetical application of an ORC system will be done to aid in determining the potential value inherent in this particular technique of waste energy utilization. The assumptions are that the unit is to be used to replace a 15 HP electric motor operating on a continuous basis, 8760 hours per year. The electric savings (in KW) are 11.19 KW per hour or 98024.4 KW per year. At a cost of $0.02 per KWH, the savings in electric cost would be $1960.00. Using the annual electric cost escalation rates given in Section 4, the projected savings in electric cost are illustrated in Table 7.1.

YEAR	ELECTRIC SAVINGS ($)
1978	2,278
1979	2,642
1980	3,065
1981	3,457
1982	3,803
1983	4,183
1984	4,601
1985	5,061
1986	5,568
1987	6,124
1988	6,737
1989	7,411
1990	8,152

Table 7.1. Electric cost savings using ORC as a replacement for a 15 HP electric motor.

A present sum analysis is illustrated in Figure 7.2 and demonstrates a payback period of 4 1/2 years. The analysis was carried out using the savings data from Table 7.1, $11,500 initial capital outlay, $0.1 per year maintenance cost and a 7% rate of return. It is to be emphasized that this analysis does not include the cooling and heating supply piping system and component cost or the installation cost.

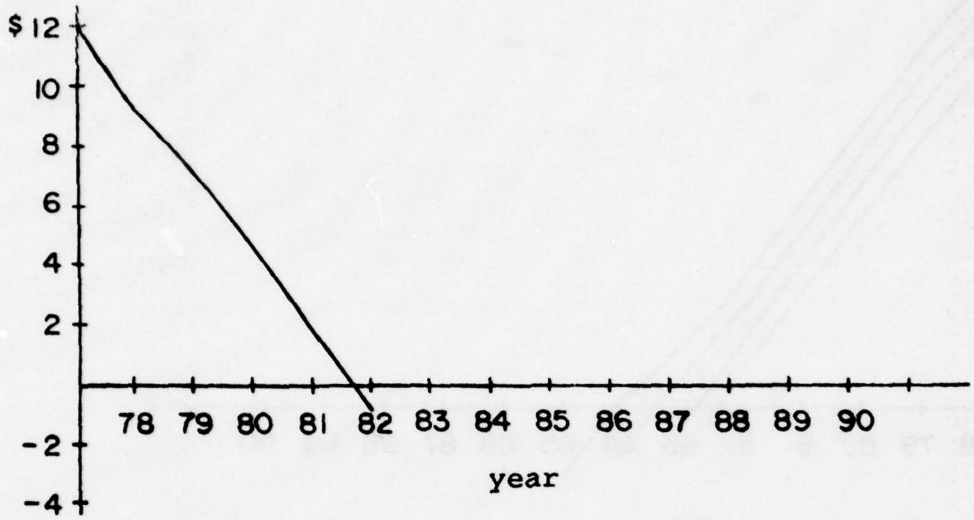

Figure 7.2. Present sum analysis for replacing a 15 HP electric motor with an ORC system. Note that horizontal axis labelling corresponds to the end of each year.

15 HP is the output of the Organic Rankine Cycle System, and the unit could be used in steam and HTHW heating plants as a prime mover for condensate, circulating, and feedwater pumps or for compressors and other devices usually requiring an electric motor. Figure 7.3 illustrates various payback times for capital expenditures of $13,500, $14,500, $15,500, and $16,500 and is intended to reflect the increase in payback times as initial outlay for cooling and heat supply equipment increases over and above the cost of the ORC unit.

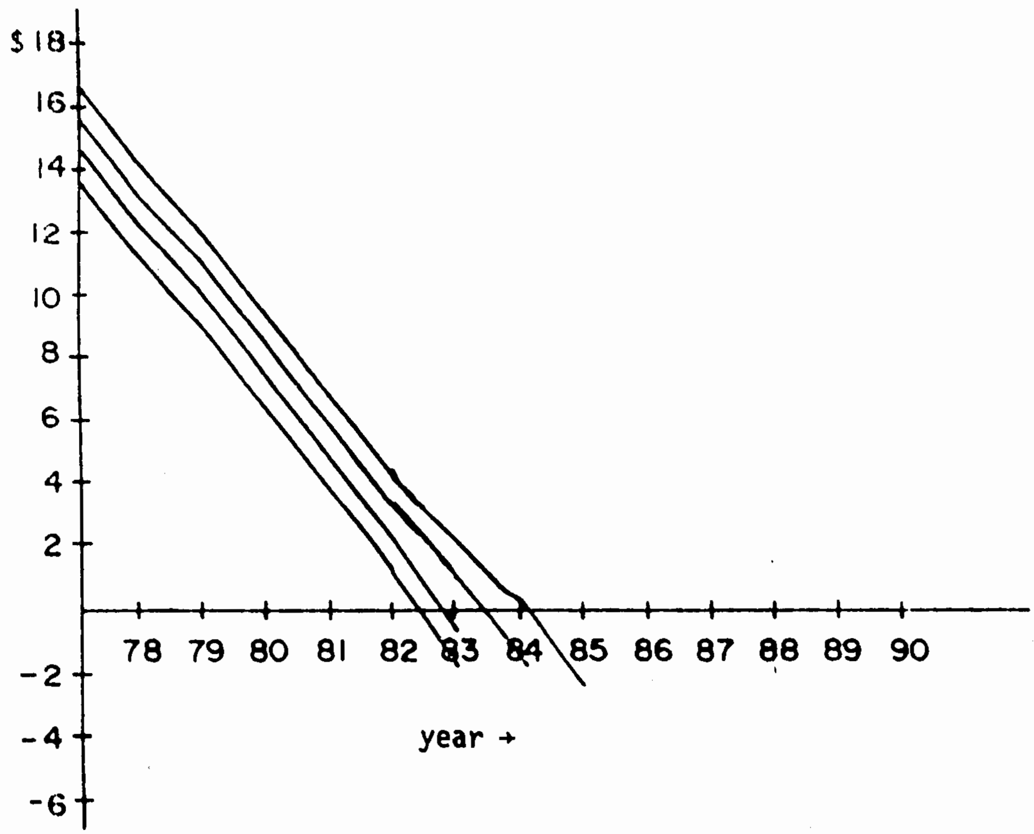

Figure 7.3. Present sum analysis indicating effect of higher initial outlay for ancillary equipment for ORC system.

As electric utility rates continue to escalate with time, the Organic Rankine Cycle power system driven by a waste heat source seems to offer a viable and economically justifiable means for waste heat recovery. Currently these units appear to be economically feasible only where a year-round source is available. In addition, stack energy flow rates should be a minimum of 350,000 BTU/hr* for suitable implementation of the 10 KW unit. The end use should be clearly identified before implementation of the unit.

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*If the stack gas temperature were reduced from 600°F to 300°F, the required stack gas flow rate for 350,000 BTU/hr is 5000 lbm/hr or 1000 cfm, which is low by industry-wide comparison.

8. ENERGY RECOVERY IN COMPRESSED AIR SYSTEMS

8.1 Lowering the Output Pressure of Compressed Air Systems

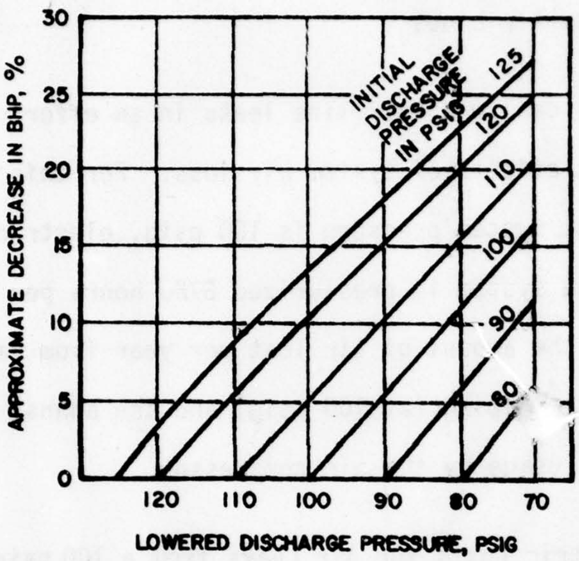
Many air compressors in operation at Air Force installations have been in service for an extended period of time. It is possible that in some cases, machines may be supplying air at pressures and flow rates that are no longer needed, because of changes in mission or mission requirements. If service requirements can be met at lower supply pressures, substantial savings in electric costs may be realized.

The first step to this approach would involve a site survey to determine if air pressure can be lowered. If, for instance, nine out of ten outlets could function satisfactorily with lower supply pressure, then a small local air compressor adequate for the one outlet with greater need may be feasible. In addition, a check with unit manufacturers should be carried out to insure that a given compressor pressure can be lowered and the degree of difficulty associated with the process.

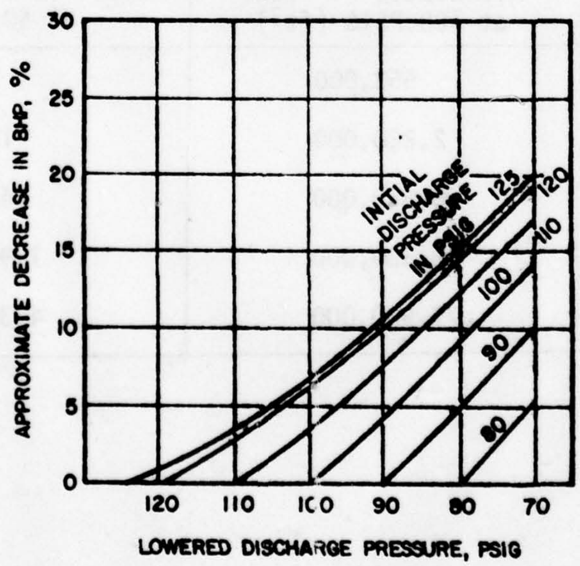
To illustrate the benefits that may be expected from lowering air pressure, an analysis will be done concerning a two-stage reciprocating air compressor located in Area B, Building 42 at WPAFB. This unit is driven by a 200 HP electric motor and supplies field air at 100 psig. For the sake of analysis, it will be assumed that the air pressure can be reduced to 85 psig and still provide adequate service to all outlets. In addition, the unit operates continuously. The number of hours will be assumed to be 8760 hours annually.

The approximate reduction in electric BHP requirement in reducing supply pressure from 100 to 85 psig is 7.25% (ef. Figure 8.1). The annual savings in electric demand is: 94,756 KWH/year.

At an assumed cost of $0.020 per KWH, a savings of $1933 is immediately possible. Considering projected escalation in utility rates, it is clear that reducing air compressor discharge pressure where possible can be beneficial from an energy as well as monetary standpoint.



Single Stage Reciprocating and
Rotary Screw Compressor



Two Stage Reciprocating and
Centrifugal Compressors

Figure 8.1. Pressure Reduction, BHP Reduction Relationships

8.2 Leaking Compressed Air Lines

This section will deal with air line leaks in an effort to point out long-term economic effects of system air loss. For this analysis, it will be assumed that system pressure is 100 psig, electric cost is \$0.020 per KWH, and the system is pressurized 8760 hours per year. Table 8.1 illustrates the amount of air lost per year from various diameter holes in system piping (at 100 psig) and the annual cost for associated electricity usage by the air compressor.

Table 8.1. Electric Costs for Air Leaks from a 100 psig Line for a 22 BHP Electric Prime Mover Providing 100 cfm Free Air Flow.

Hole Diameter (inches)	Air Wasted Per Year at 100 PSIG (ft^3)	Electric Cost Per Year at \$0.02/KWH
1/32	553,000	30.30
1/16	2,220,000	121.00
1/8	8,880,000	486.00
1/4	35,500,000	1,940.00
3/8	79,900,000	4,370.00

In the long run, leaks in air system piping can have significant effect on operating economics. The numbers quoted in Table 5.1 are for a single leak; consider the effect of several hundred leaks which may occur in a large distribution system.

It seems a good practice to conduct planned periodic inspection routines on air systems to insure piping and component integrity. Reliance on casual detection of leaks should be avoided due to physical qualities of air (colorless, odorless). In addition, noise associated with such leaks may easily be obscured by plant or industrial processes. One effective technique for locating system air leaks involves swabbing likely areas with a soap-water mixture. Conducting inspections after duty hours is also effective. Likely areas include points subject to mechanical and physical stress such as joints, elbows, points of passage through walls or buildings, and valves.

8.3 Energy Recovery from Large Air Compressors

In general, air exiting a compressor is at a relatively high temperature as a result of the compression process. The air must be cooled to eliminate (condense) entrained water and oil vapors, and this is usually accomplished with a water-cooled heat exchanger (after-cooler). If needed, additional air cooling is provided by an Organic Rankine Cycle "chiller" unit. In either case, the energy removed from the air is dissipated to an environment outside the structure which houses the compressor. Because the housing structure frequently has a heating requirement, it seems advantageous to use energy rejected from the compressed air for building heating.

The simplest scheme would involve placing a heat exchanger in the air stream after the compressor discharge and before the aftercooler, where air temperatures are typically on the order of 250^{o}F. Energy thus recovered could be ducted to nearby areas where a heating requirement exists, with losses minimized by proper design and insulation of the ductwork.

The most efficient heat exchanger that could be used in this instance is a heat pipe exchanging array (95+%). After contacting several manufacturers of heat pipes, it was determined that no such equipment currently exists, but several manufacturers suggested that the idea could prove marketable and envision a completed unit costing approximately $750.

The amount of energy contained in the discharge of a compressor of the type found at the Ram Air Facility, TAFB, is calculated below. The results are based on an air flow of 6 lbm/sec at 300°F.

$$Q = M\, C_p\, \Delta T \qquad (8\text{-}1)$$

where Q is the heat transfer rate, M the mass flow rate, C_p the specific heat constant pressure, and ΔT the temperature difference.

Substituting representative values

$$Q = \left(\frac{21{,}600\ \text{lbm}}{\text{hr}}\right)\left(\frac{.24\ \text{BTU}}{\text{lbm}\ F^{o}}\right)(300-70) = 1.19 \times 10^{6}\ \text{BTU/hr}$$

Removing 50% of the thermal energy in the air would result in a recovery of 596,160 BTU/hr. After heat exchange, the compressed air temperature entering the aftercooler would be reduced from 300°F to 185°F. This would result in a decreased cooling load on the aftercooler and chiller unit equal to the same value.

The energy removed from the compressed air by the heat pipe array could be sent to specific sites as required through use of ductwork and a blower. The air flow rate for the heating side of the array is anticipated to be about twice that of the compressed air side. This is to insure that the delivery temperature will be in a safe and comfortable range. Under stated conditions, the delivery air temperature for 100% efficient heat recovery is 127°F.

Assuming that building heating is currently being supplied by local gas-fired heaters, the annual gas savings can be calculated as shown below. The analysis is based on 75% efficient heaters and on an annual usage of four months, continuous.

$$\text{Annual Gas Savings} = 596{,}160\ \frac{\text{BTU}}{\text{hr}} \times \frac{2880\ \text{hrs}}{\text{year}} \times \frac{1\ \text{ft}^3}{1020\ \text{BTU}} \times 1.33$$

$$= 2{,}244{,}367\ \text{ft}^3/\text{year}$$

At a cost of \$1.36 per MCF, this fuel savings would equal \$3052. Electric energy costs associated with the blower must be subtracted from fuel savings. The amount of air flow the blower must provide is 10,683 CFM based on an air density of 0.0674 lbm/ft^3 for the given conditions. A horsepower rating of 15 HP for the electric motor should be adequate. Annual electric costs for blower operation are shown below and are based on 2¢ per KWH.

$$\text{Annual Electric Costs} = 15\ \text{HP} \times \frac{.746\ \text{KW}}{\text{HP}} \times \frac{2880\ \text{hrs}}{\text{year}} \times \frac{\$0.02}{\text{KWH}}$$

$$= \$644.54$$

$$\text{Net Savings} = \$3052 - \$644.54$$

$$= \$2407.46$$

A savings analysis is presented in Table 8.2, and is based on a natural gas escalation rate given in Section 4.

YEAR	FUEL SAVINGS	ELECTRIC COST	NET SAVINGS
1978	$3510	$748	$2762
1979	4036	867	3169
1980	4642	1006	3636
1981	5013	1076	3937
1982	5414	1152	4262
1983	5847	1232	4615
1984	6315	1319	5001
1985	6820	1411	5409

Table 8.2 Net dollar savings for heat recovery from an air compressor using heat pipe array.

The present sum analysis for this concept is illustrated in Figure 8.2, and is based on a heat exchanger cost of $750, ancillary equipment cost of $2500, installation cost of $1500, maintenance costs of $100 per year, and savings as outlined in Table 8.2.

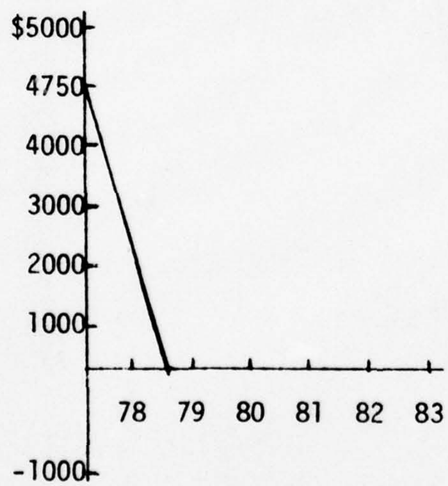


Figure 8.2. Present sum breakeven point for heat pipe heat exchanger.

Fuel savings will be a function of:

a) Utilization of the recovery device (length of heating season).

b) Compressed air exit temperature and flow rate. Dollar savings will of course be related to the type of heat provided, i.e., steam, forced

air electric, etc.

From the analysis presented, and in light of the rapid payback, it is evident that energy recovery from compressed air for structural heating is feasible from an economic as well as an energy savings viewpoint.

These sections have presented several concepts for energy conservation and recovery with regard to air compressors; conservation measures should precede energy recovery. Organic Rankine Cycle power units are usually not practical with air compressor units, because the energy available is usually too low. The units discussed here are larger than one may expect to find at most facilities.

9. WASTE ENERGY RECOVERY FROM REFRIGERATION AND AIR CONDITIONING SYSTEMS

9.1 Basic Considerations for Energy Recovery

Refrigeration and air conditioning systems can generally be represented by a four-stage process involving a compressor, condenser, expansion valve, and evaporator. The compressor and expansion valve effect changes in working fluid temperature and pressure, while the condenser and evaporator function primarily as heat transfer devices.

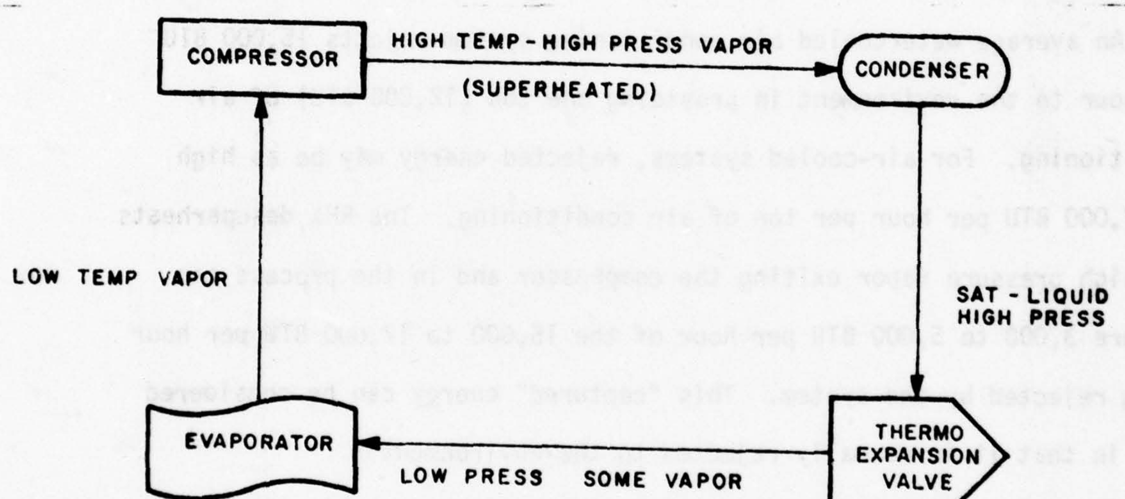


Figure 9.1. Basic air conditioning and refrigeration process.

Energy recovery techniques as applied to refrigeration and air conditioning systems use a heat exchanger placed between the discharge of the compressor and the condenser. Working fluid leaving the compressor is a high temperature, high pressure (superheated) vapor. The recovery heat exchanger (RHX) is designed to function as a desuperheater for the

working fluid prior to its entry into the condenser. The RHX should be designed so condensation occurs only in the condenser and not in the RHX itself, to avoid premature lowering of system pressure. In addition, the RHX must offer minimal restriction to fluid flow to prevent an excessive pressure drop in the working fluid as it passes through the device. Counterflow heat exchangers have proved the most efficient for this particular application, because of maximized temperature differentials, hence optional heat transfer rates. Axial counterflow heat exchangers are best for small ($\leq$ 12 ton) units, while a shell and tube type should be used for larger applications.

An average watercooled air conditioning system rejects 15,000 BTU per hour to the environment in providing one ton (12,000 BTU) of air conditioning. For air-cooled systems, rejected energy may be as high as 17,000 BTU per hour per ton of air conditioning. The RHX desuperheats the high pressure vapor exiting the compressor and in the process can capture 3,000 to 5,000 BTU per hour of the 15,000 to 17,000 BTU per hour being rejected by the system. This "captured" energy can be considered free in that it is normally rejected to the environment.

The most common use for energy recovery systems of this nature involve heating water. The basic process is illustrated schematically in Figure 9.1. Hot water can be used for space heating, or can be used in other hot water applications (i.e., potable water, rinse water for industrial processes). If, for example, a 50 gallon hot water tank is used in a heat recovery scheme involving a 5 ton air conditioning system, the maximum amount of heat that can be recovered is 25,000 BTU per hour. This is sufficient to raise the 50 gallons of water from 70^{o}F to 130^{o}F in one hour.

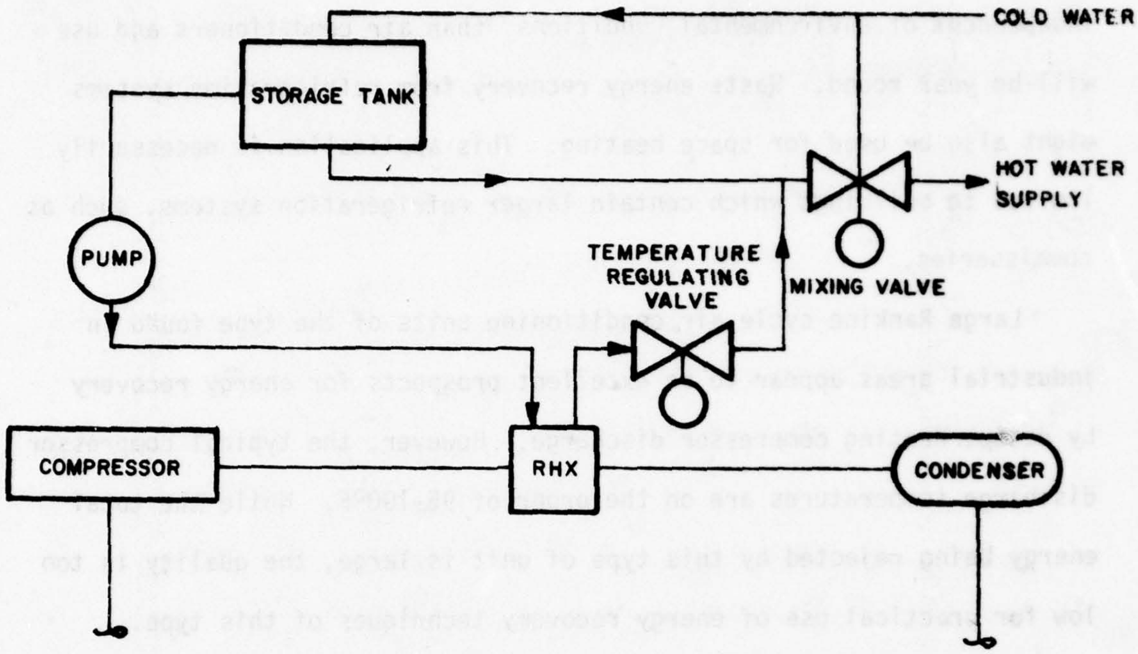


Figure 9.2. Heat recovery from air conditioning and refrigeration

Twenty-five thousand BTU per hour is the equivalent of 7.32 KW of electrical energy or about 33 cubic feet of natural gas in a 75% efficient water heater. The savings per hour are insignificant when electrical costs are around 2 cents per KWH and gas costs are roughly $2.00 per MCF. The key to the waste energy concept lies in utilization. Small differences in large numbers over a long period of time can add up to substantial savings Preliminary studies conducted at Patrick AFB, Florida (Reference 6) with actual hardware indicate significant savings. Clearly, geographic and climactic conditions are reflected directly in use of air conditioning systems, hence heat recovery capabilities. Refrigeration systems are more

independent of environmental conditions than air conditioners and use will be year round. Waste energy recovery from refrigeration systems might also be used for space heating. This application is necessarily limited to buildings which contain larger refrigeration systems, such as commissaries.

Large Rankine cycle air conditioning units of the type found in industrial areas appear to be excellent prospects for energy recovery by desuperheating compressor discharge. However, the typical compressor discharge temperatures are on the order of 95-100°F. While the total energy being rejected by this type of unit is large, the quality is too low for practical use of energy recovery techniques of this type.

Water heating by air conditioning units is possible, but is only effective if the air conditioning unit is substantially used. In addition, the low compressor discharge temperature of some units makes water heating impractical. Use of the air conditioning unit, compressor discharge temperature and location of potential end use should all be carefully determined. Refrigeration units which normally operate year round, have higher discharge temperatures and have a potable water heating requirement nearby are more likely prospects.

9.2 Refrigeration Heat Recovery

While the system described in Section 9.1 can operate year round, the concept presented in this section limits energy recovery to seasons when heating is required and is limited to refrigeration systems. The concept centers around use of a second condenser which acts as a heater unit. It can be placed in ductwork or be a separate unit. When a requirement exists for building heating, a circuit is energized which bypasses the primary condenser diverting the hot compressor discharge to the heat recovery condenser where the gas gives up its heat to ventilating air. At the same time, the primary condenser, fans, and water pumps which are usually located on the building exterior are de-energized. The partially condensed working fluid is then sent to the primary condenser to re-instate normal flow conditions. The hot gas temperature is nominally at 150°F so that the amount of heat transfer is significant. Depending on local climatic conditions, this approach can be expected to provide 60% to 100% of the heating requirement for large installations.

The concept is easily expanded where multiple refrigeration units exist by simply adding an array of secondary condensers in the ventilation ductwork. This insures mechanical independence and continuity of operation in the event of repairs to a single refrigeration system.

The total amount of heat recoverable from all refrigeration units within a given structure should be capable of providing at least half the heating requirements. In addition, only medium temperature units of at least 5 HP or low temperature units of 7 1/2 HP or above should be considered. The heating savings must be weighed against the cost of diverting valves, refrigerant piping, condensing unit, and ancillary equipment for each refrigeration unit. Payback periods of two to three years, however, are not uncommon.

Selection of proper recovery hardware is a function of many variables including building size, occupant load, ambient conditions, ventilation requirements, structural materials, refrigeration unit size and number, and geographical location. Each installation must be considered individually due to the wide variability in pertinent parameters. For this reason, a recovery analysis will be carried out for a hypothetical situation based on general assumptions. A detailed analysis would require accurate information to serve as a data base before realistic recovery capabilities and subsequent capital outlays could be determined. Therefore, the analysis done here is intended to be generally representative of the concept of recovery from refrigeration systems.

It will be assumed that the "store" contains: 1) a 15 HP dairy case operating at a suction temperature of 10°F; 2) a 10 HP ice cream case operating at a suction temperature of -35°F; 3) a 15 HP frozen foods case operating at -35°F. The amount of heat rejected from each unit in BTU/hr is 145,000, 56,000, and 69,500 BTU per hour respectively. The total rejected energy available for store heating is 270,500 BTU/hr.

Assuming that heat is provided by natural gas at 75% efficiency, annual gas savings realizable by utilization of refrigeration heat recovery is 353 cubic feet per hour or 1,015,807 cubic feet per year based on 24 hours per day, 4 months per year of continuous utilization. Associated fuel cost savings are listed in Table 9.1. Gas costs are assumed to be $1.36 per MCF.

Year	Fuel Cost Savings
1978	1,589
1979	1,827
1980	2,101
1981	2,269
1982	2,451
1983	2,647
1984	2,858
1985	3,087
1986	3,334

Table 9.1 Fuel Cost Savings for Use of Refrigeration Heat Recovery

The particular heat recovery unit under consideration is Model #4CF 1206H, manufactured by Tyler Refrigeration Corporation of Niles, Michigan, and is intended to be placed directly in existing HVAC ductwork. Typical unit cost is approximately $1500, installation costs are given as $500 (based on 15 hours labor @ $10/hr and piping costs of $3.50 per linear foot for 100 feet of piping) and negligible maintenance costs.

Associated savings are as listed in Table 9.1. The present sum analysis is shown in Figure 9.2.

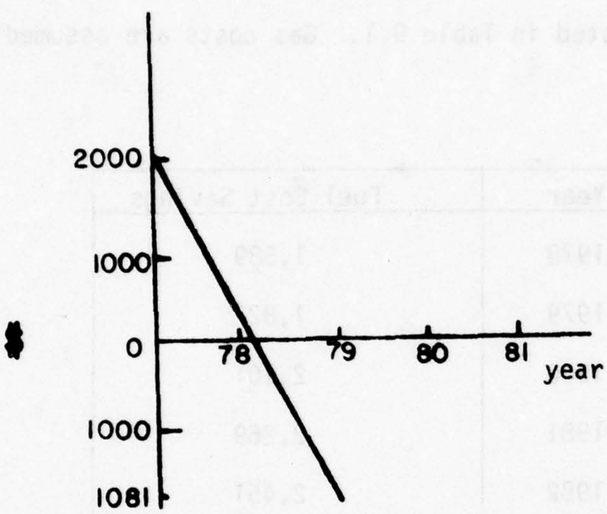


Figure 9.3. Present sum analysis for heat recovery from refrigeration units.

Indicate that capital recovery occurs quite rapidly, provided that the "store" could use all this energy.

An appropriate location for such a unit is in areas where infiltration loss is significant, such as an unloading dock area. Such a unit is located at the Wright-Patterson Air Force Base Commissary. It is designed to provide heating for the unloading dock-storage area in the winter months.

Nearly every Air Force base has a commissary and implementation of such units could significantly reduce the building heating load. Care

must be taken to insure that the heat recovery unit is not oversized for the particular unit. If this occurs, then the unit will become overloaded and either compressor failure or reduced cooling will result.

10. WASTE ENERGY RECOVERY SCHEME FOR THE LAUNDRY FACILITY AT WPAFB

Laundry facilities represent an excellent source of low quality waste energy that can be suitably used for space heating. The laundry facility at WPAFB has been analyzed as an example. This is in a 33,000 ft^2 building (Building A-745) near a steam plant (Building A-770). The total heat requirement for the facility is provided by the steam plant and includes steam for: 1) space heating, 2) hot water (for washing machines), and 3) clothes dryers. Space heating is done by forced air to steam-coil heat exchange and the same technique is used for clothes drying. Hot water is produced by immersing steam coils in water stored in several large tanks. In all cases, condensate return is practiced. A small amount of steam is vented to the atmosphere, but is thought by supervisory personnel to be insignificant heat loss. There are no fuel fired processes within the facility and air conditioning is non-existent. Ventilation supply is provided by window fans, with roof mounted exhaust fans.

After some investigation, it has been determined that a major area where heat recovery might be economically feasible involves the clothes dryers. Within the building are seven large dryers manufactured by the American Dryer Corporation of Boston, Massachusetts. Each unit exhausts air to the atmosphere at about 105°F and 3000 CFM. Each unit

has its own exhaust ductwork which is 16 inches in diameter and uninsulated. On the average, all seven units are in operation six hours per day, five days per week, 52 weeks per year. Drying air is exhausted rather than recycled due to the relatively high moisture content from the drying process. Supply air for dryer heat exchanges is taken from within the building at 3000 CFM per unit.

The primary heat recovery scheme involving the facility dryers deals with heat exchange between dryer exhaust and intake. Clearly, the quality of the source (105°F) is limited, but the quantity (3000 CFM) is significant.

After investigation, the range of choice for recovery equipment was reduced to two "best" alternatives; an enthalpy wheel and a heat pipe array. An enthalpy wheel extracts both latent and sensible heat from the source stream; which is desirable for efficiency. However, a percentage of the moisture removed by the wheel will be transferred to the incoming air stream by evaporation. The problem becomes apparent when the equipment mission is considered; dryers are designed to remove moisture from garments. If the incoming air contains more moisture than ambient, drying time will be extended proportionally. For this reason, the heat pipe array is the appropriate means to satisfactory heat recovery, because it can recover and transfer both latent and sensible heat. In addition, heat pipes are static devices, having no moving parts and requiring no external power source.

Since the heat pipe array will be removing latent heat, some condensation will occur, and for this reason, a drain pan type of arrangement must be included in the overall plan. Condensation can be piped to the existing drainage system which should not involve significant problems.

One additional recommendation must be mentioned before beginning detailed analysis, and this concerns dryer air supply. Present practice is to "pull in" air from the building itself for dryer supply. Obviously, the infiltration rate is directly related to the amount of air removed by the dryers which, in turn, is directly related to heating requirements during the heating season. Therefore, the practice of using conditioned air from within the building should be discontinued. Instead, outside air can be used by installing intake ductwork.

The advantages associated with the heat exchange between exhaust and supply air streams are twofold. First, the amount of steam required for space heating should decrease because of reduced infiltration of relatively cold outside air. Secondly, preheating incoming air should reduce the amount of steam required for clothes drying, if the air can be heated to a temperature above normal inside ambient. Since the dryer supply is currently taken from within the building (which is assumed to have an indoor design temperature of $70^{o}F$), the heat exchanging process must raise the outside air intake temperature to greater than $70^{o}F$ before steam savings will be realized. Steam savings are directly proportional to intake air temperature, and hence, the degree of preheating the heat pipe array can provide. The heat exchanging configuration is illustrated in Figure 10.1.

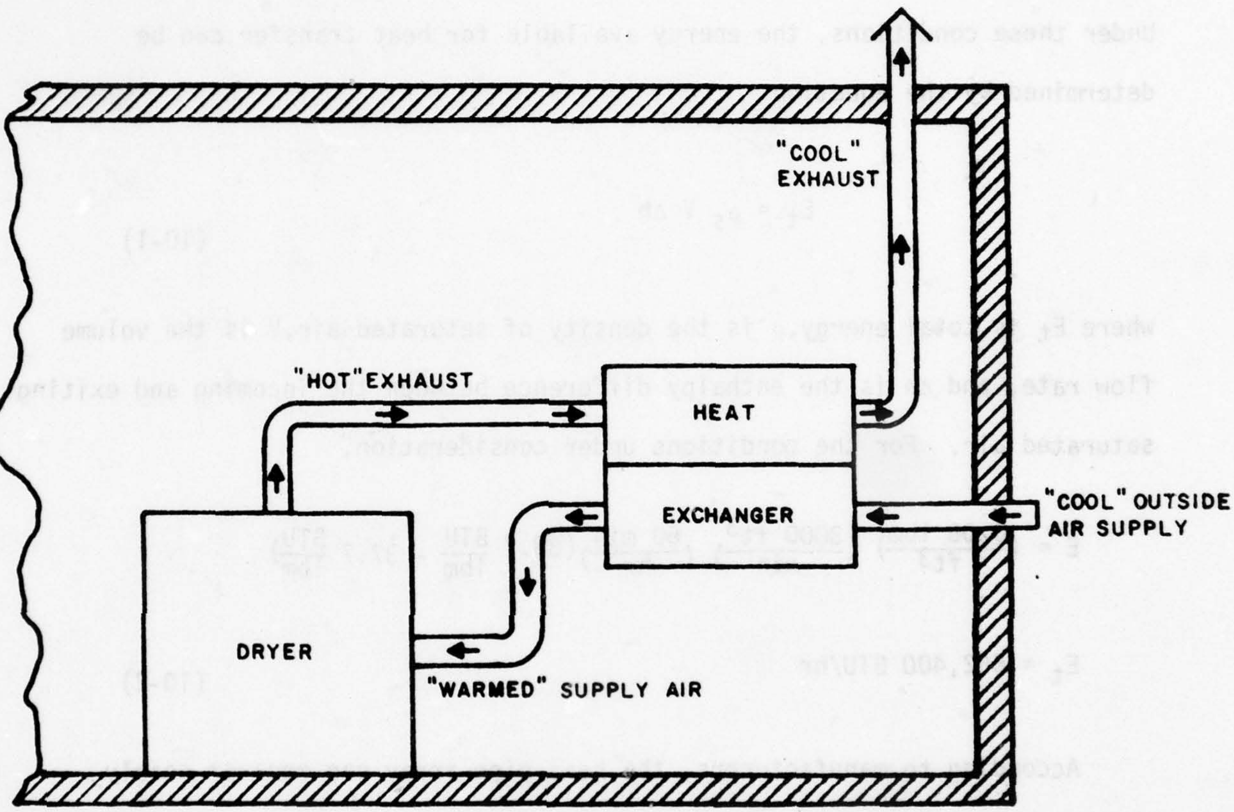


Figure 10.1. Heat Exchange Configuration for the Laundry Facility at WPAFB.

It has been determined that the heat pipe array under consideration can heat incoming outside air from 50°F (yearly average outdoor temperature) to 83°F. In the process, layer exhaust air is reduced from 105°F to 73°F. Under these conditions, the energy available for heat transfer can be determined by the equation

$$E_t = \rho_s \, V \, \Delta h \qquad (10\text{-}1)$$

where E_t is total energy, ρ is the density of saturated air, V is the volume flow rate, and Δh is the enthalpy difference between the incoming and exiting saturated air. For the conditions under consideration,

$$E = \left(\frac{0.08 \text{ lbm}}{\text{ft}^3}\right)\left(\frac{3000 \text{ ft}^3}{\text{min}}\right)\left(\frac{60 \text{ min}}{\text{hr}}\right)\left(83.7 \frac{\text{BTU}}{\text{lbm}} - 37.7 \frac{\text{BTU}}{\text{lbm}}\right)$$

$$E_t = 662{,}400 \text{ BTU/hr} \qquad (10\text{-}2)$$

According to manufacturers, the heat pipe array can recover nearly all of the sensible energy and approximately 60% of the latent energy, in the exhaust. The sensible energy is a fraction of the total energy available for transfer and is determined by

$$E_s = \rho_s \, V \, C_p \, \Delta T \qquad (10\text{-}3)$$

where E_s is the sensible energy, ρ_d the density of dry air, C_p the specific heat at constant pressure for dry air, and ΔT the temperature difference.

$$E_s = \left(\frac{.076\ \text{lbm}}{\text{ft}^3}\right)\left(\frac{3000\ \text{ft}^3}{\text{min}}\right)\left(\frac{60\ \text{min}}{\text{hr}}\right)\left(\frac{.24\ \text{BTU}}{\text{lbm}\ \text{F}^\circ}\right)(105^\circ\text{F} - 73^\circ\text{F})$$

$$E_s = 105{,}062\ \text{BTU/hr} \tag{10-4}$$

The amount of energy that can be recovered by the heat pipe array includes all of the sensible energy ($\dot{E}_s$) and 60% of the latent energy. Therefore, the total energy that can be transferred from the exhaust to the supply air under these conditions is

$$Q = E_s + .60\ (E_t - E_s)$$

$$Q = 105{,}062 + (.6)\ (557338)$$

$$Q = 439{,}465\ \text{BTU/hr} \tag{10-5}$$

However, the amount of energy that is saved for a single dryer is

$$E_n = \left(\frac{.076\ \text{lbm}}{\text{ft}^3}\right)\left(\frac{3000\ \text{ft}^3}{\text{min}}\right)\left(\frac{60\ \text{min}}{\text{hr}}\right)\left(\frac{.24}{\text{lbm}\ \text{F}^\circ}\right)(82^\circ\text{F} - 50^\circ\text{F})$$

$$E_n = 105{,}062\ \text{BTU/hr} \tag{10-6}$$

Therefore, the energy in the exhaust of a single dryer can provide adequate recovery for four units. The exhaust from two units, ducted into a single heat pipe array, will provide enough energy to preheat intake air for all seven dryers. The heat pipe array can be

partitioned to accommodate the unbalanced flow rates between the intake and exhaust sections of the unit. The exhaust from the five remaining units would be rejected to the environment in the normal manner.

The first manifestation of fuel savings will result from reduced steam demand for each dryer due to preheating the intake air. The total reduction in energy usage due to preheating is 735,535 BTU/hr at stated conditions. Annual energy savings are given by

$$Q_s = (735{,}434 \ \frac{\text{BTU}}{\text{hr}}) \ (\frac{6 \text{ hrs}}{\text{day}}) \ (\frac{5 \text{ days}}{\text{week}}) \ (\frac{52 \text{ weeks}}{\text{year}})$$

$$Q_s = 1.15 \times 10^9 \text{ BTU/year} \qquad (10\text{-}7)$$

The second aspect of energy savings will be due to reduced infiltration that will occur by supplying the dryers with outside air. Each dryer extracts 3000 CFM from the structure for a total of 21,000 CFM or 1,260,000 cubic feet per hour. The building volume is 660,000 cubic feet, so that there are 1.91 air changes per hour. Infiltrating air is outside air that must be heated during colder months to maintain a comfortable working environment of 70°F. The amount of heat that must be added to infiltrating air is proportional to outside temperature and will vary accordingly. At 40°F, average daily outside temperature will be assumed (based on 90 year averages) for the heating months of November through March. The amount of heat that must be added by the space heating steam coils is

$$Q = Mc_p \, \Delta t$$

$$Q = 1{,}260{,}000 \, \frac{ft^3}{hr} \times \frac{0.0720 \text{ lbm}}{ft^3} \times \frac{.24 \text{ BTU}}{\text{lbm } F^o} \times (70\text{-}40)$$

$$= 653{,}184 \, \frac{BTU}{hr} \tag{10-8}$$

The seven dryers operate six hours per day, five days per week, 52 weeks per year. But the heating season extends from November through March and in light of this fact, the total infiltration losses per year are

$$Q = 653{,}184 \, \frac{BTU}{hr} \times \frac{6 \text{ hrs}}{day} \times \frac{105 \text{ days}}{year}$$

$$= 4.115 \times 10^8 \tag{10-9}$$

The total energy savings for the laundry facility by reducing overall steam demand will be 1.562×10^9 BTU/yr. Savings for different possible fuels will be calculated for relative comparison and are based on a 75% efficient heating process (boiler).

If the boilers are coal fired, the amount of coal savings are:

$$S_c = 1.562 \times 10^9 \, \frac{BTU}{yr} \times 1.33 \times \frac{\text{lbm}}{14{,}550 \text{ BTU}} \times \frac{1 \text{ ton}}{2000 \text{ lbm}}$$

$$S_c = 71.4 \text{ tons/year} \tag{10-10}$$

The analysis is based on the high heating value for bituminous coal. At a cost of \$31.44 per ton, this represents a 1977 annual saving of \$2245. A savings schedule for coal is shown in Table 10.1 for 1978 through 1990 using the escalation rates of Section 4.

If the boilers are fired with #2 (diesel) fuel oil, the annual oil savings potential amounts to

$$S_o = 1.562 \times 10^9 \frac{BTU}{yr} \times 1.33 \times \frac{gallon}{144{,}000\ BTU}$$

$$S_o = 14.427 \times 10^3 \text{ gallons/year} \qquad (11\text{-}11)$$

This represents a yearly savings of \$5049 assuming a cost of \$0.35 per gallon of fuel.

If the boilers are heated with natural gas, the annual gas savings is

$$S_g = 1.562 \times 10^9 \frac{BTU}{yr} \times 1.33 \times \frac{ft^3}{1080\ BTU}$$

$$S_g = 1.924 \times 10^6 \text{ Ft}^3\text{/year} \qquad (11\text{-}12)$$

At an assumed cost of \$1.36 per MCF, the savings potential will be \$2,500.

The recovery unit is assumed to cost \$8,250. Total installation costs will be assumed to be \$2,500 with annual maintenance costs assumed to be \$210 which will be for cleaning the finned array.

Year	Savings
1978	$ 2,470
1979	$ 2,716
1980	$ 2,988
1981	$ 3,137
1982	$ 3,294
1983	$ 3,459
1984	$ 3,632
1985	$ 3,814
1986	$ 4,004
1987	$ 4,204
1988	$ 4,415
1989	$ 4,635
1990	$ 4,867

Table 10.1. Fuel Cost Savings for Coal Using Heat Pipe Recovery Unit at WPAFB Laundry Facility.

The present sum analysis for the coal-fired boilers is shown graphically in Figure 10.2.

The breakeven point on the graph is shown to be slightly under four years.

If the boilers are fired with #2 (diesel) fuel oil, the annual fuel savings will be 14,427 gallons per year. The 1977 cost is $0.35 per gallon with 16% escalation in rates until 1980 and 8% thereafter. Table 10.2 reflects savings over a 13 year period. The time of installation is January 1, 1978.

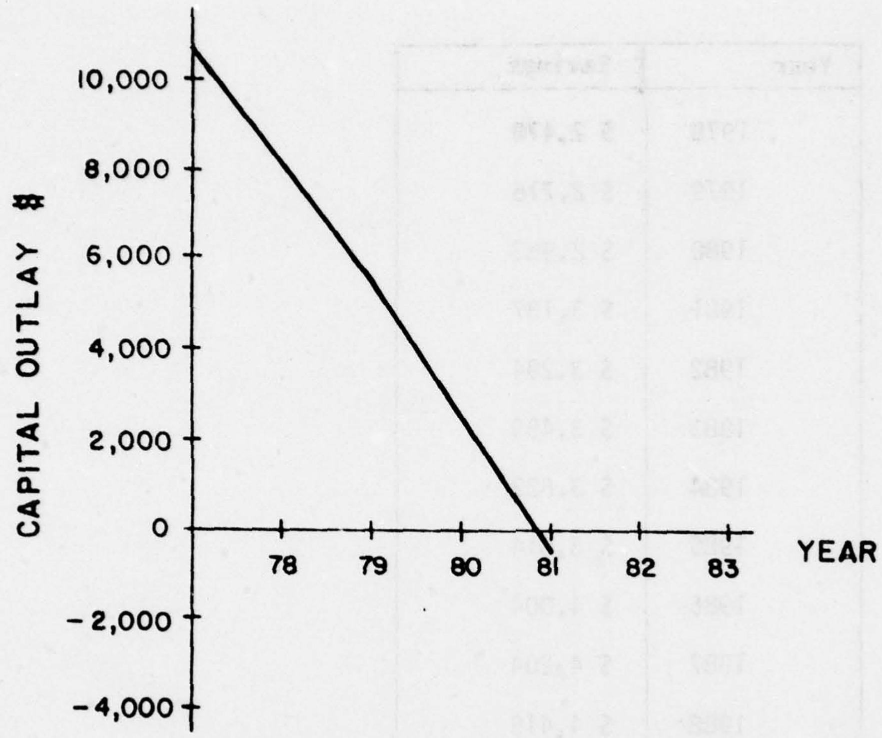


Figure 10.2 Present sum analysis for coal-fired boilers.

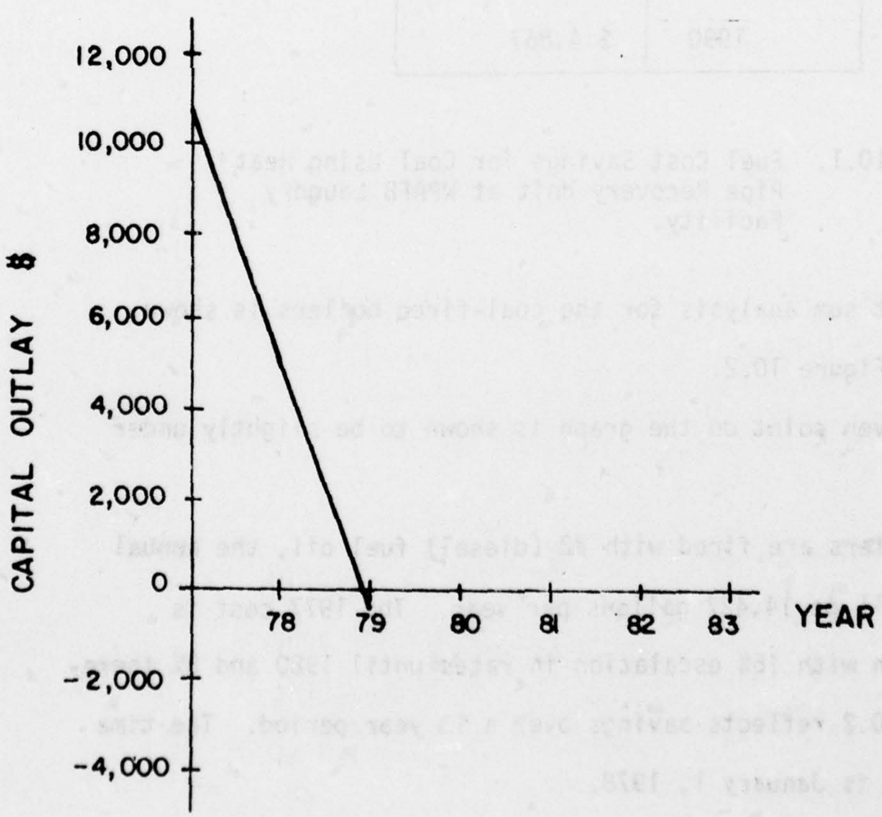


Figure 10.3 Present sum analysis for oil-fired boilers.

Year	Savings
1978	$ 5,857
1979	$ 6,794
1980	$ 7,881
1981	$ 8,511
1982	$ 9,192
1983	$ 9,928
1984	$10,722
1985	$11,580
1986	$12,506
1987	$13,507
1988	$14,587
1989	$15,754
1990	$17,014

Table 10.2. Fuel Cost Savings for #2 Fuel Oil

The break-even point as shown in Figure 10.3 is approximately two years.

If the boilers are fired with natural gas, the annual fuel savings will amount to 1.924 cubic feet. 1977 costs are about $1.30 per MCF and are expected to escalate at 15% until 1980 and 8% thereafter. Table 10.3 shows projected fuel cost savings anticipated through the use of the Heat Pipe array.

Year	Savings
1978	$ 2,876
1979	$ 3,308
1980	$ 3,804
1981	$ 4,108
1982	$ 4,437
1983	$ 4,792
1984	$ 5,175
1985	$ 5,589
1986	$ 6,036
1987	$ 6,519
1988	$ 7,040
1989	$ 7,604
1990	$ 8,212

Table 10.3. Fuel Cost Savings for Natural Gas

Figure 10.4 illustrates the payback time for natural gas fired boilers as slightly under five years.

It is clear from the analysis that the installation of Heat Pipe heat exchangers at the Laundry Facility at WPAFB is sound from an economic as well as an energy standpoint. The analysis of this particular facility is generally representative of all similar installations. The multiple fuel approach is specifically intended to illustrate wider applicability and it is hoped that it will fulfill its purpose.

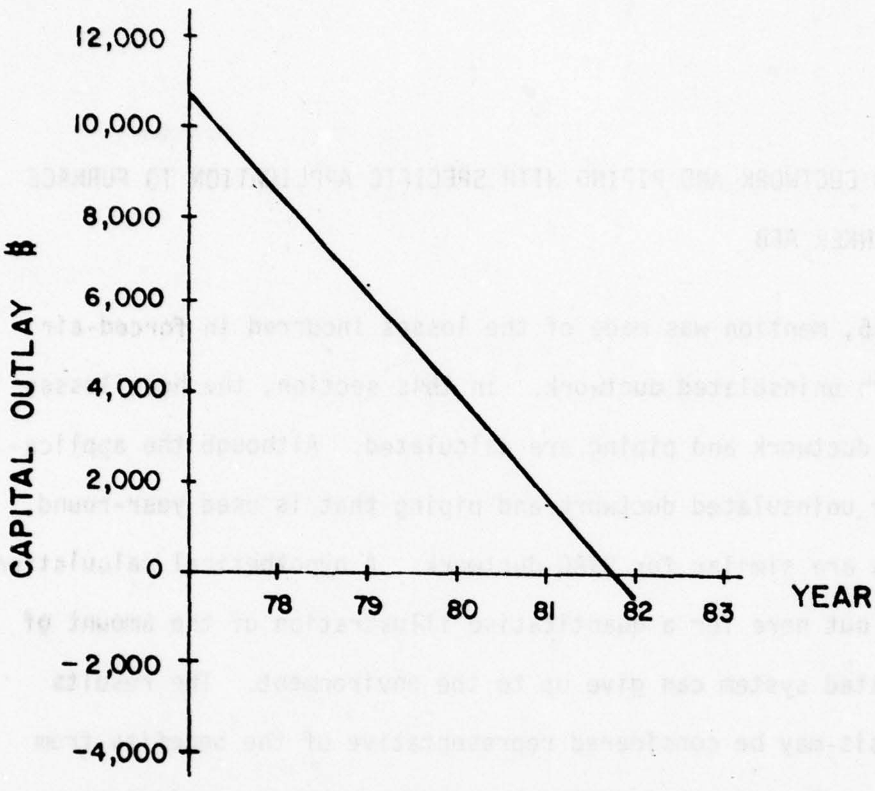


Figure 10.4. Present sum analysis for natural gas-fired furnaces.

11. UNINSULATED DUCTWORK AND PIPING WITH SPECIFIC APPLICATION TO FURNACE UNITS AT TINKER AFB

In Section 5, mention was made of the losses incurred in forced-air gas furnaces with uninsulated ductwork. In this section, the heat losses associated with ductwork and piping are calculated. Although the application here is for uninsulated ductwork and piping that is used year-round, the computations are similar for HVAC ductwork. A hypothetical calculation will be carried out here for a quantitative illustration of the amount of heat an uninsulated system can give up to the environment. The results from this analysis may be considered representative of the benefits from insulating "exposed" piping and ductwork, and may serve as an indicator for cost benefit analysis associated with insulation of such systems.

The factors which determine heat transfer from uninsulated ductwork and piping are: flow rate of fluids within the system, temperature of fluid relative to environment, surface area available for heat transfer, physical length of system, system construction material, type of fluid, and orientation (vertical, horizontal) of system piping. Calculation of heat transfer rates from piping surfaces due to each mode can be time consuming; therefore, tables have been constructed to aid the analyst. The 1972 ASHRAE Handbook of Fundamentals (Reference 9) is suggested for this use.

For example, consider the yearly heat loss from a 40 foot run of 4 inch steel pipe carrying flue gases at 400^{o}F, operating for 2000 hours per year.

For an 80^{o}F ambient temperature

$$\Delta T = 400^{o}F - 80^{o}F = 320^{o}F$$

Using Table 11, page 371 of the ASHRAE Handbook, the energy loss coefficient is

$$U = 3.45 \frac{BTU}{hr\ ft^2\ F^o}$$

$$Q = (3.45 \frac{BTU}{hr\ ft^2 F^o})\ (40\ ft)\ \pi\ (.375\ ft)\ (320\ F^o)$$

$$Q = 52,000\ BTU/hr$$

The yearly loss would then be

$$Q = 1.04 \times 10^8\ BTU$$

Energy gained in this manner is attractive in the winter months, but in the summer the air handling and/or air conditioning system must absorb this load. In most cases, however, industrial areas require air handling and/or air conditioning units that operate year round to control the excess heating associated with the equipment.

Relating this to the amount of cooling (in tons) that the air handling equipment must provide to remove the heat input from the uninsulated piping,

$$Q = 1.04 \times 10^8 \times \frac{\text{BTU}}{2000 \text{ hr}} \times \frac{1 \text{ hr Ton}}{12{,}000 \text{ BTU}}$$

$$Q_{out} = 8{,}665 \text{ tons}$$

Consider the use of two inches of mineral fiber (slag or glass) for insulation. This particular material has a thermal conductivity of 0.45 at 400°F. The appropriate equation for this calculation is the radial heat transfer equation.

$$q_s = \frac{T_o - T_a}{\dfrac{r_s \ln (r_s/r_o)}{k} + R_s} \tag{11-1}$$

where

q_s = rate of heat transfer per square foot at outer surface of insulation, BTU/ft^2.

k = thermal conductivity of insulation, $\text{BTU/hr-ft}^2\text{-F}^o\text{-inch}$.

T_a = ambient temperature

T_o = temperature at inner surface of insulation

r_s = outer radius of insulation, inches.

r_o = inner radius of insulation, inches.

R_s = surface resistance $\text{hr-ft}^2\text{-F}^o\text{/BTU}$.

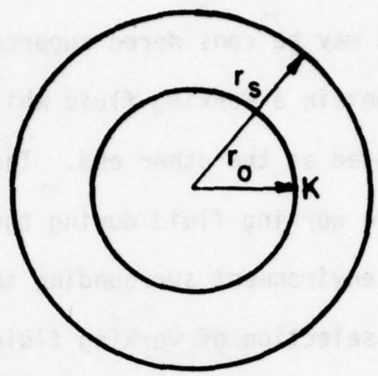


Figure 11.1. Circular piping nomenclature

Using the values from the ASHRAE Handbook

$$q_s = \frac{320\ F^o}{\frac{4\ \ln(4/2)}{0.36} + .65}$$

$$q_s = 38\ BTU/hr\ ft^2$$

On a yearly basis,

$$Q = 4.77 \times 10^6\ BTU/year$$

This figure represents a 95% reduction in the heat added to the building environment by insulating the system as specified. Modern technology permits reduced loading on air conditioning (via insulation), while still allowing piping system heat to be used, if desired. Isothermal devices called heat pipes that operate with various types of working fluid have been perfected to the degree that they are readily available

on the market. Heat pipes may be considered superconductors for energy transfer as heat. They contain a working fluid which is evaporated at one end of the pipe and condensed at the other end. The latent heat of vaporization is given up by the working fluid during the condensation process, and is transferred to the environment surrounding the condensing end of the heat pipe. By proper selection of working fluid and internal pressure, virtually any temperature range and heat transfer rate can be achieved.

Isothermics Incorporated of Augusta, New Jersey manufactures a small heat pipe array that is designed to be installed in ductwork systems with diameters between 6 and 9 inches. When placed in hot gas streams (400 to 800 degrees Farenheit) these fan-forced units can extract 5000 to 15000 BTU/hr and deliver air at flow rates of 80 to 120 SCFM, at temperatures ranging between 140 to 200°F. They are adjustable to thermostatic control and can be activated by a local temperature sensor if automatic operation is desired. Up to 20 feet of ducting may be connected to an outlet of the heat pipe array to transport warmest air to a different location, without appreciable loss of efficiency.

The sermetal paint treating shop located in building 3001, Tinker AFB, is one area where small heat pipe arrays could prove beneficial. This shop contains six 120 KW electric ovens designed to operate at 650°F for use in the heat treating segment of the sermetal process. The ovens are ventilated by natural convection, and each is connected to a common exhaust header by pipework approximately 4 inches in diameter. The exhaust header eventually exits through the roof after a 50 foot horizontal run.

A common practice of the operators in the sermetal shop is to preheat parts prior to heat treatment, by opening the oven doors and placing the parts to be preheated directly in the opening. This results in significant losses of warmed air from the furnace interior, as the preheat process requires about 12 minutes per article. A suitable alternative to this means of preheating would be to recover energy from the exhaust system and transfer it to a preheat enclosure as shown in Figure 11.2. A small heat pipe array would be ideally suited for this process, as no cross contamination occurs between the exhaust gases and the warmed air.

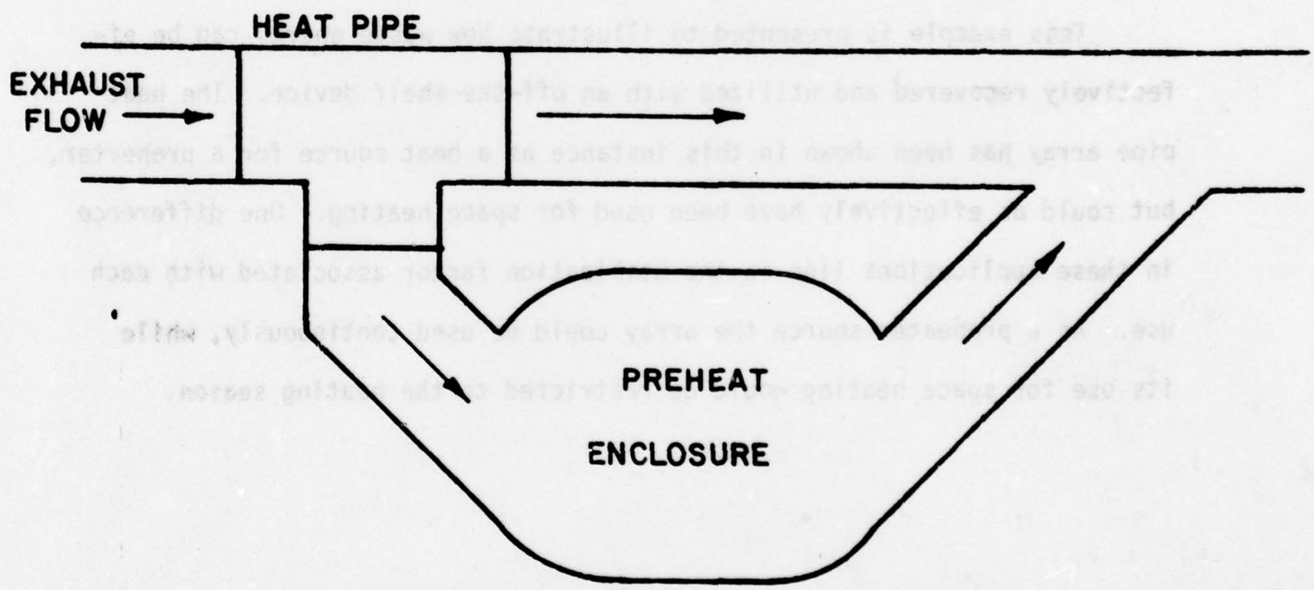


Figure 11.2. Example for use of a heat pipe heat exchanger on small diameter piping and ductwork.

If a preheat enclosure were used, the time required to heat a one-half cubic foot volume of stainless steel from 75°F (ambient) to 150°F for pre-

heating purposes would be about 30 minutes for a 5000 BTU/hr heat pipe array. Unit cost for the heat pipe device is listed by the manufacturer at $170 each. Quantity discounts are available.

The payback period for an investment in a small heat pipe unit is a function of unit cost, preheat enclosure cost, associated ductwork, installation costs, and energy savings. A capital outlay of approximately $500 would cover these costs. Electric energy savings on the order of 500 watts per hour can be realized by altering the preheat procedure as described and will amount to first year dollar savings of about $90 at $0.02 per KWH. The payback period for the installation would be about 5 years.

This example is presented to illustrate how waste energy can be effectively recovered and utilized with an off-the-shelf device. The heat pipe array has been shown in this instance as a heat source for a preheater, but could as effectively have been used for space heating. One difference in these applications lies in the utilization factor associated with each use. As a preheater source the array could be used continuously, while its use for space heating would be restricted to the heating season.

PART III

12. THE RAM AIR TEST FACILITY AT TINKER AFB

12.1 General Considerations

The Ram Air Facility located in Building 210, Tinker Air Force Base, Oklahoma City, Oklahoma has a unique mission. Air driven turbine generator sets from various aircraft are tested under conditions approximating as closely as possible those found aboard operating aircraft. Each unit under test is supplied with high pressure (300 psig) air, the temperature of which can be regulated between ambient and 1000^{o}F, through mixing of ambient and heated air as desired. Electrical loading of the units is accomplished with resistive load absorbing banks which dissipate electrical energy as heat to the environment. Several discrete resistor configurations are available to cause the desired amount of load to be "seen" by the turbine generators.

High pressure air is provided by some combination of six, 3-stage reciprocating compressors, each driven by an 800 HP synchronous electric motor. Interstage air cooling is provided to maintain mechanical parameters of the compressors, and an aftercooler is situated in line with each compressor to remove moisture and oil entrained in the air. The relatively dry compressed air passes into an accumulator and then to the main air header. A 60-ton rankine cycle chiller unit is placed in the main air stream to complete moisture and oil removal and insure dry air for turbine operation. The main air header separates downstream of the chiller unit to form the cold and hot high pressure air headers. Varying amounts of air designated for the hot air header are heated in

two of three gas fired air heaters which raise the air temperature from ambient to 1000°F. Each heater is designed to handle an air flow rate of 6 lbm/sec, and under these conditions uses 6770 cubic feet of natural gas per hour.

Hot and cold air are blended in each test cell to attain the desired air temperature. Air exiting the turbine is exhausted to the roof area along with the cooling air used to maintain the test cell environment. A constant bleed orifice is installed into the hot air header to insure a continuous flow of air and preclude condensation in the header. The orifice exhausts 0.541 lbm/sec to the outside environment through a roof vent. The basic energy transfer process for the Ram Air Facility is shown in Figure 12.1, while the associated idealized thermodynamic diagram is shown in Figure 12.2.

Considering one compressor and its associated internal cooling as an isolated system, the specific energy transfer is

$$\text{work in} = W_{in} = 800 \text{ HP}$$

$$\text{air mass flow rate} = M = 6 \text{ lbm/sec} = 360 \text{ lbm/min}$$

$$Q_{out} = M\, C_p \{T_{in} - T_{out}\} + W_{in} \qquad (12\text{-}1)$$

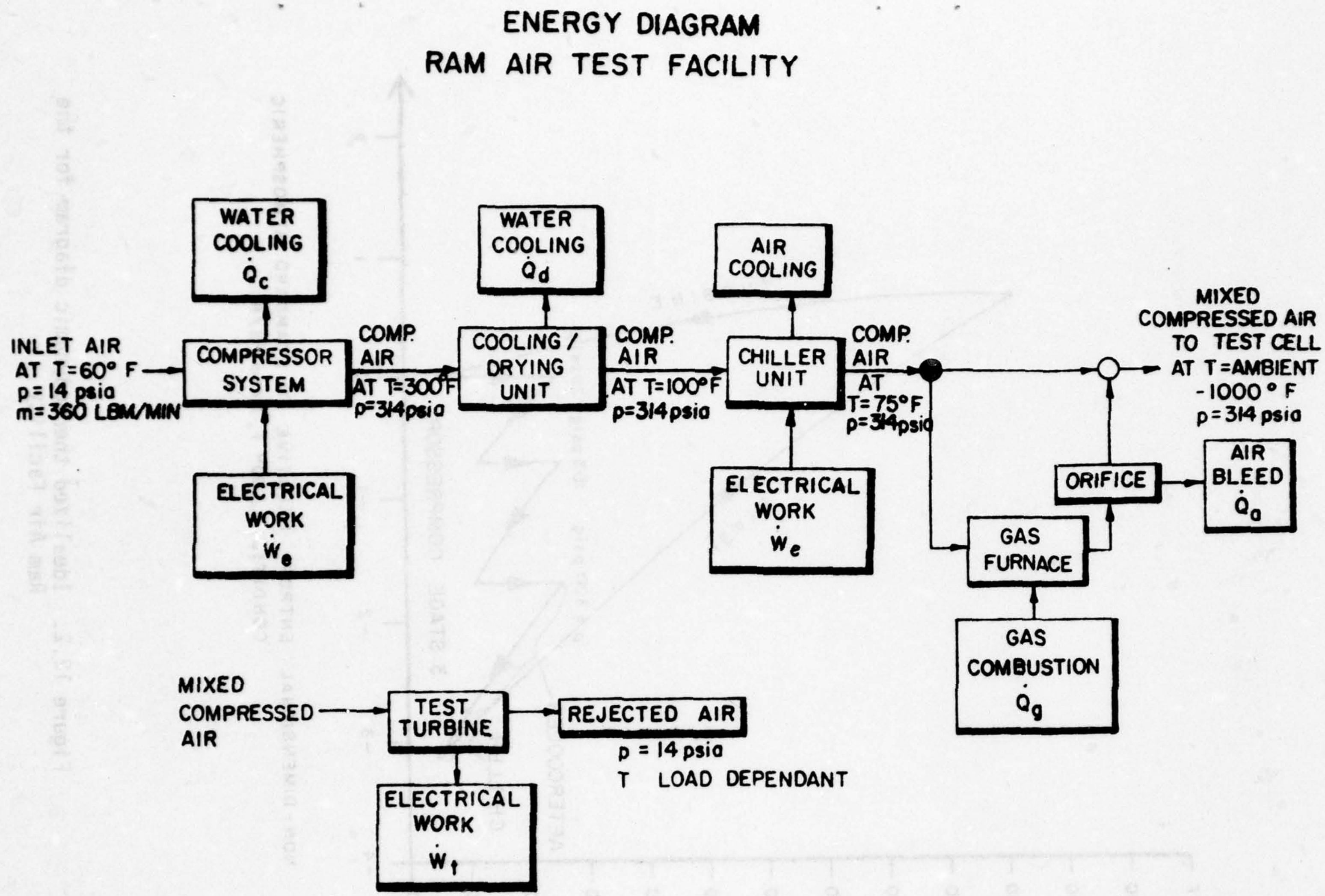


Figure 12.1 Energy diagram for Ram Air Test Facility

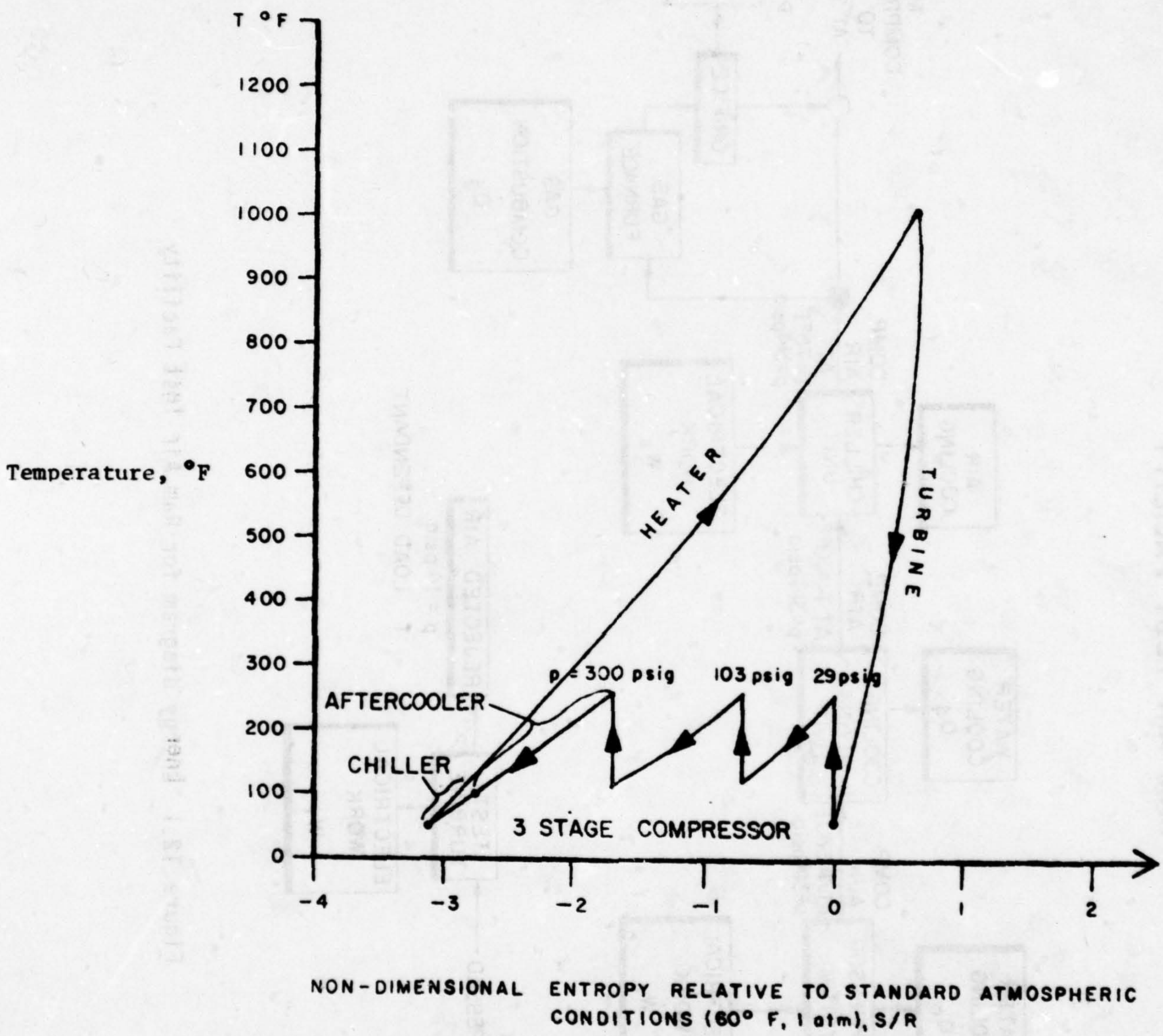


Figure 12.2. Idealized thermodynamic diagram for the Ram Air Facility

where Q_{out} is the heat flux due to cooling, C_p the specific heat at constant pressure for air, and T the air temperature. Substituting present conditions,

$$Q_{out} = (360 \frac{lbm}{min}) (\frac{.24\ BTU}{lbm\ F^o})(60-300)\ F^o + 800\ HP$$

$$Q_{out} = 311\ HP$$

The calculation reveals that 311 HP of the 800 HP electrical (high quality) input for the compression process is removed as low quality heat, while the remaining 489 HP appears as an increase in the enthalpy of the compressed air. The energy removed in cooling the compressed air and compressor (311 HP, 8×10^5 BTU/hr) is dissipated to the environment by cooling towers and a circulating water system. While the total amount of energy thus removed is large, the quality is relatively low because cooling water exit temperature is in the 100^oF to 110^oF range.

The aftercooler located downstream of the third stage compressor outlet reduces the compressed air temperature from 300^oF to 100^oF for the preliminary removal of entrained moisture and oil. The aftercooler is supplied by circulating water which enters at about 70^oF and exits at 100^oF to 110^oF. The total amount of energy removed from the aftercooler is about one million BTU per hour including the release of latent heat from condensation of water vapor.

After being cooled in the aftercooler, the compressed air passes to an accumulator which acts as a buffer to cushion impulses from the reciprocating compressor. Thus, the air pressure is nearly a constant 300 psig.

Compressed air then enters the 10-inch main header and passes through the 60 ton chiller heat exchanger. The air temperature is further reduced in the chiller from 100°F to 75°F to insure that the air is as dry as possible for air turbine operation. The total amount of energy removed by the chiller is proportional to air flow which is in turn related to the number of compressors in operation, and the degree of loading of each. Assuming a 6 lbm/sec (360 lbm/min) air flow rate, the energy removed by the chiller is:

$$Q_{out} = (360 \frac{lbm}{min}) (\frac{.24\ BTU}{lbm\ F^{o}})(25\ F^{o}) (\frac{60\ min}{hr})$$

$$Q_{out} = 1.3 \times 10^{5}\ BTU/hr$$

The average number of compressors in operation is four. Therefore, the energy removed is 5.2×10^{5} BTU per hour, corresponding to 43.2 tons of cooling. The amount of electrical energy necessary at 72% load on the chiller is about 63% of the rating of the motor or approximately 38 KW based on a coefficient of performance of 4.

The main air header splits downstream of the chiller forming the cold and hot air manifolds. Compressed air to be heated enters two of three gas-fired heaters. As stated previously, each heater can heat 6 lbm/sec of air from 75°F to 1000°F, consuming 6770 cubic feet of natural gas per hour in the process. The design efficiency of the heater is 72%. Translating

this figure to actual energy added to the compressed air yields the following result for each furnace.

$$Q_{comb} = 6770\ \frac{ft^3}{hr} \times \frac{1080\ BTU}{ft^3} \times .72$$

$$Q_{comb} = 5.26 \times 10^6\ BTU/hr$$

The exhaust flue temperature is about 350^oF and the associated energy loss is in the neighborhood of 2×10^6 BTU/hr. The units are designed to preheat combustion air.

The orifice plate mentioned earlier results in a heat loss from the system of:

$$Q_a = (32.46\ \frac{lbm}{min})\ (\frac{.24\ BTU}{lbm\ \ F^o})\ (930\ F^o)$$

$$Q_a = 4.35 \times 10^5\ BTU/hr$$

The primary areas of interest from an energy recovery viewpoint involve utilization of: 1) cooling water effluent, 2) high temperature compressor discharge, 3) chiller replacement, 4) combustion exhaust, 5) electric load absorber replacement, 6) electric motor replacement, and 7) hot air vent orifice. Each area will be subject to separate investigation beginning with the analysis of the use of cooling water effluent.

12.2 Rejected Energy Recovery at the Ram Air Test Facility

The amount of energy removed from the compressor and compressed air is about 1.8×10^6 BTU/hr for each compressor. In the cooling process, circulating water temperature increases from 70°F to 105°F. The mass flow rate is calculated to be approximately 440 gallons per minute for each compressor.

Use of a recuperative heat exchanger supplied by cooling water effluent and placed downstream of the chiller unit in the hot air line could re-heat the compressed air exiting the chiller. If it is assumed that the air could be heated to 100°F, the total amount of energy that the recuperative heat exchanger could add to the compressed air would be:

$$Q = (720\ \frac{\text{lbm}}{\text{min}})\ (\frac{.24\ \text{BTU}}{\text{lbm F}^{\text{o}}})\ (25\ \text{F}^{\text{o}})$$

$$Q = 259{,}200\ \text{BTU/hr}$$

The Ram Air Facility is currently operated on a 24 hour per day, 264 day per year basis. The annual energy savings that can be realized through use of a recuperative heat exchanger as specified is therefore 1.64×10^9 BTU.

The heating value for natural gas typically distributed in the Oklahoma City area is 1080 BTU per cubic foot, but because the gas heaters

are 72% efficient, the actual amount of natural gas that the recuperative heat exchanger can save will be:

$$\text{Gas savings} = (1.64 \times 10^9 \frac{\text{BTU}}{\text{year}}) (\frac{\text{ft}^3}{1080 \text{ BTU}}) (1.33)$$

$$\text{Gas savings} = 2.11 \times 10^6 \text{ cubic feet/year}$$

At an average 1977 gas cost of $1.36 per MCF, the associated savings would be $2886.

Figure 12.3 is a cash flow diagram illustrating savings for the recuperative heat exchanger. The analysis is based upon use of a Fin-Tube heat exchanger costing about $920 with a yearly maintenance cost estimated to be $100. The heat exchanger will cause an air pressure drop of 4 psig and a 3 psig water pressure drop at 50 gpm. Installation cost is estimated to be $3,000. Gas rates are expected to escalate at 15% per year until 1980 and 8% thereafter. Figure 12.4 illustrates the results of the present sum analysis for the recuperative heat exchanger and is based on a figure of 7% for the cost of money.

The second area where energy recovery may prove possible involves use of the high temperature (300°F) compressor discharge. The amount of heat energy removed from the third stage compressor discharge air is:

$$Q = (360 \frac{\text{lbm}}{\text{min}}) (\frac{.24 \text{ BTU}}{\text{lbm F}^\circ})(300\text{-}100^\circ\text{F})$$

$$Q = 1.0 \times 10^6 \text{ BTU/hr}$$

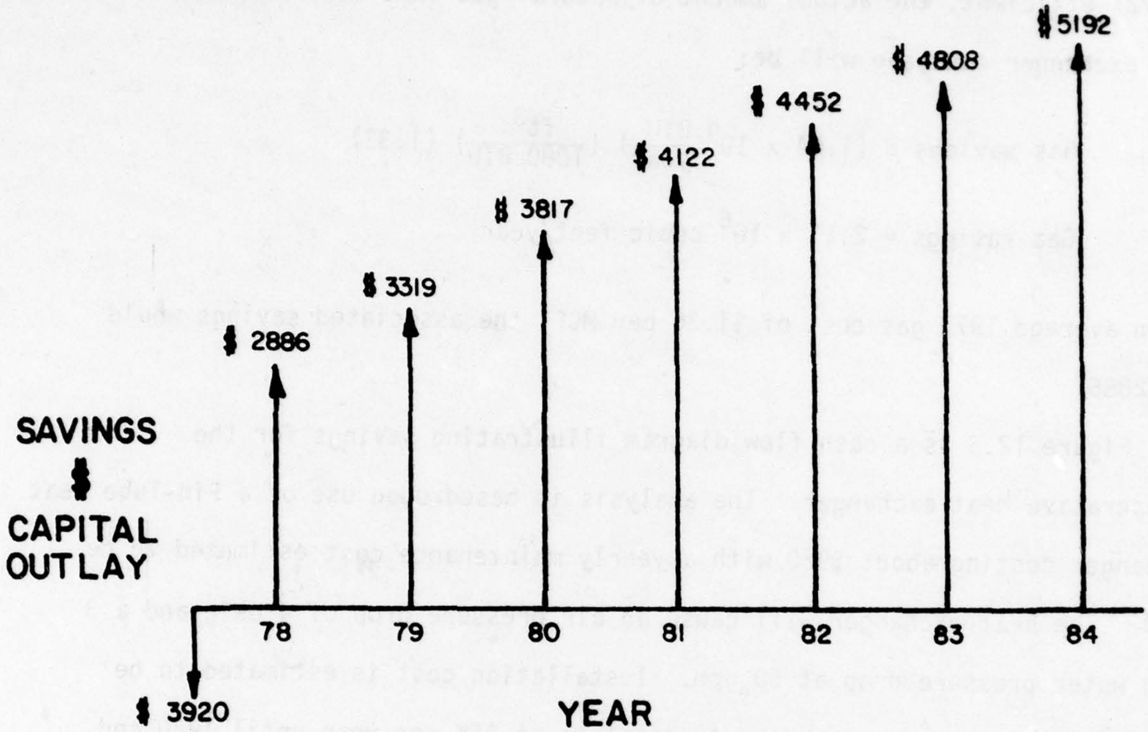


Figure 12.3. Cash flow diagram for Fin-Tube heat exchanger.

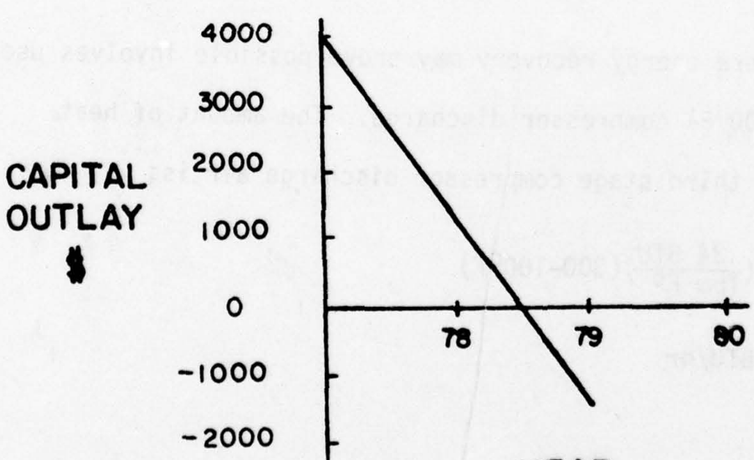


Figure 12.4. Present sum payback analysis for Brown Fin-Tube recuperative heat exchanger.

It is clear that this represents a relatively high quality, high quantity source. The most advantageous use of this energy would involve the production of shaft work, thereby converting the high quality heat source to mechanical work with a potential for producing energy.

The most suitable method for effecting energy conversion in this case incorporates an Organic Rankine Cycle (ORC) process. Heat exchange between compressed air and the organic working fluid could be accomplished by an in-line device similar to the type used for aftercooling. Assuming that a heat exchanger would be 40% efficient, the amount of energy that could be recovered from the compressed air is:

$$Q_r = (.4)(1.0 \times 10^6 \text{ BTU/hr})$$
$$= 4.16 \times 10^5 \text{ BTU/hr}$$

Given a 15% efficiency for the ORC process, the unit could produce:

$$W = (.15)(416{,}000 \text{ BTU/hr})$$
$$= 62{,}400 \text{ BTU/hr}$$
$$= 18 \text{ KW}$$
$$= 24.5 \text{ HP}$$

The scheme as outlined would require a heat exchanger in the compressed air stream for each compressor, with appropriate valving at the ORC unit to permit use of a given exchanger as required. An alternate approach might entail a central heat exchanging unit with the outlet of each compressor ducted to a central recovery unit. This would permit use

of a larger ORC unit. With an average of four compressors in operation on a continuous basis, the recovery potential is 72 KW or 98 HP.

A manufacturer of smaller ORC units is Sun Power Systems of Miami, Florida. The standard 15 HP unit (10 KW) costs about $12,000 and a specially manufactured 88 KW unit would cost approximately $96,000. Using an installation cost of $50,000 including ancillary system piping, the total installed cost for the unit would be $146,000, not including the required electrical switching equipment. Balanced against this capital outlay would be dollar savings accruing from electric energy cost reductions based on 4.56×10^5 KW per year. The electrical cost savings for 1978, for example, would amount to $9,124 @2¢ per KWH. Even for projected escalation rates of 16% until 1980 and 7% thereafter, the payback time is well in excess of 5 years. Therefore, it does not appear that the conceptual scheme as outlined is financially practical at present.

A third area of interest from an energy conservation standpoint involves replacement of the 60 ton chiller unit with a desiccant/filtration system. Air filtration technology has improved considerably in recent years and units are now available that have the capability for sub-micron absolute filtration. These units are specifically designed for compressed air systems and operate at temperatures up to 120 degrees Fahrenheit.

Associated energy savings would manifest as: 1) a reduction in electrical usage, 2) reduced natural gas consumption resulting from elimination of the 75°F to 100°F air heating requirement.

Recall that a 72% average load on the chiller unit requires 38 KW of electrical energy per hour for operation. Based on 24 hours per day, 264 days per year, the annual electrical consumption for the chiller is 241,000 KWH per year or 8.22×10^8 BTU/yr. This figure represents the amount of electrical energy that can be saved by elimination of the chiller unit.

The second aspect of savings would result from elimination of the need for the natural gas fired air heaters to reheat the compressed air flow 75°F to 100°F. At a maximum air flow rate of 6 lbm/sec per compressor, for four compressors, the energy savings potential is:

$$Q = (1440 \frac{\text{lbm}}{\text{min}}) (\frac{.24 \text{ BTU}}{\text{lbm F°}})(100\text{-}75°\text{F})$$

$$Q = 518{,}400 \text{ BTU/hr}$$

Because the air heaters are 72% efficient, the amount of natural gas saved by elimination of the chiller unit is:

$$\text{Gas Savings} = (\frac{518{,}400 \text{ BTU}}{\text{hr}})(\frac{\text{ft}^3}{1080 \text{ BTU}})(1.39)$$

$$\text{Gas Savings} = 667 \text{ cubic feet per hour}$$

$$\text{Annual Savings} = (\frac{667 \text{ ft}^3}{\text{hr}})(\frac{24 \text{ hrs}}{\text{day}})(\frac{264 \text{ days}}{\text{yr}})$$

$$= 4.23 \times 10^6 \text{ ft}^3/\text{year}$$

The proposed filtration system to replace the chiller unit will incorporate: 1) a one-micron filter, 2) a micro-fine filter for oil removal, 3) a desiccant air dryer for final moisture removal. An additional one-micron filter will be needed downstream from the air dryer to remove any particulates introduced during the drying process. The desiccant system will require 15% of the total dried air flow for use in regeneration of the desiccant. This does not cause an additional energy load because of the need to supply air well in excess of the required flow rate to maintain stable pressures for the turbine inlet conditions. Instead of being exhausted, the excess air will assist in the drying operation.

The total annual savings combine the savings from electrical and natural gas reductions and amount to 4.1×10^9 BTU per year, 8.2×10^8 BTU/year from electrical savings and 3.3×10^9 BTU/year from gas savings. Figure 12.5 is the cash flow diagram associated with installation of the filter/desiccant system and is based on natural gas of $1.366 per MCF and $0.02 per KWH for electricity. Figure 12.6 is the present sum analysis for the equipment. The break-even point is based on $25,000 first cost, $15,000 installation cost, and $1000 annual maintenance costs. As illustrated, the present sum break-even time is about four years based on the stated assumptions.

Another area of interest involves use of flue gases from the air heater combustion process. Two of three installed air heaters are in operation on a continuous basis. Under full load conditions each furnace uses 6770 cubic feet of natural gas per hour. The combustion temperature

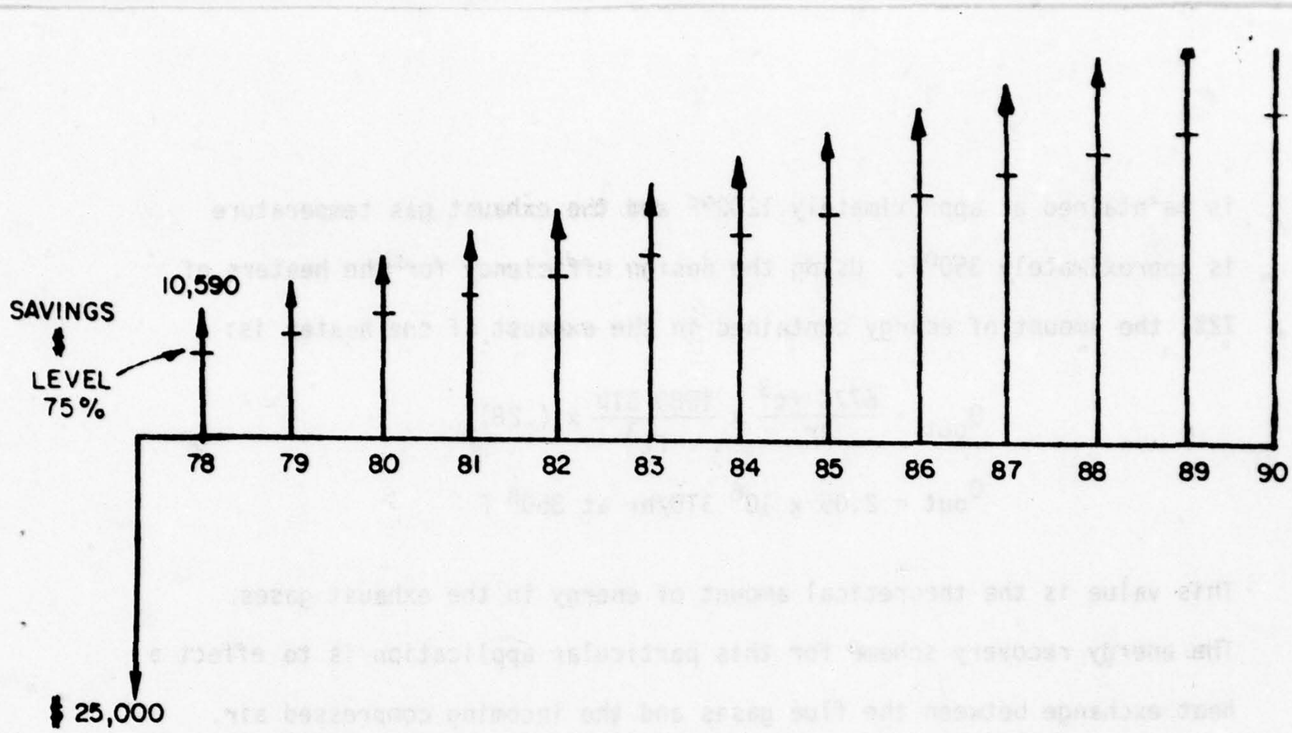


Figure 12.5 Cash flow diagram for desiccant air drying system.

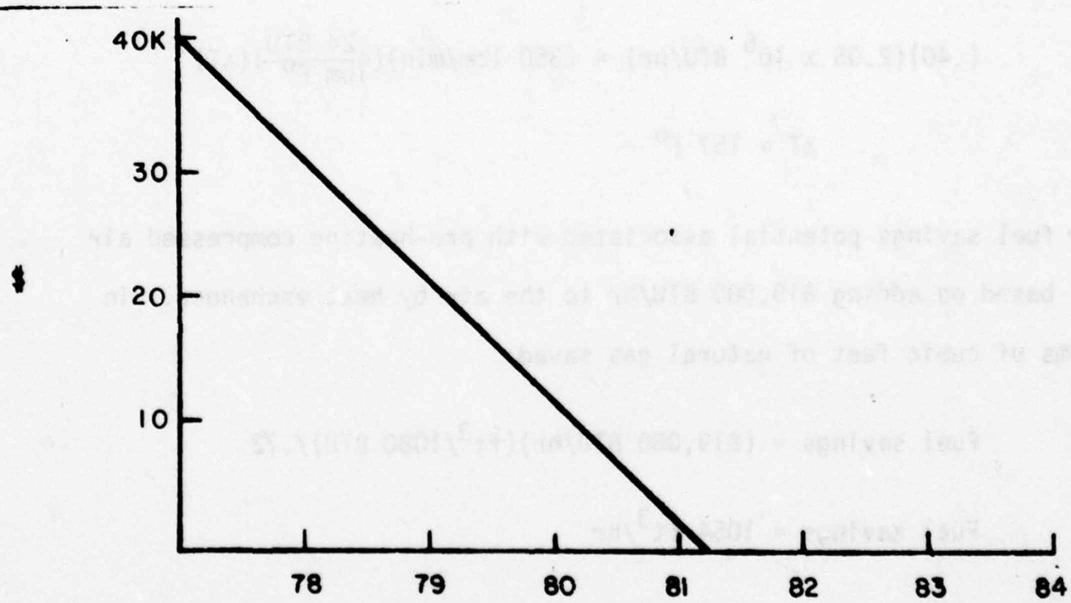


Figure 12.6 Present sum analysis for desiccant air drying system.

is maintained at approximately 1200°F and the exhaust gas temperature is approximately 350°F. Using the design efficiency for the heaters of 72%, the amount of energy contained in the exhaust of one heater is:

$$Q_{out} = \frac{6770\ ft^3}{hr} \times \frac{1080\ BTU}{ft^3} \times (.28)$$

$$Q_{out} = 2.05 \times 10^6\ BTU/hr\ at\ 350^\circ\ F$$

This value is the theoretical amount of energy in the exhaust gases. The energy recovery scheme for this particular application is to effect a heat exchange between the flue gases and the incoming compressed air.

For a gas-to-air heat exchanger that is 40% efficient, the expected temperature rise is:

$$(.40)(2.05 \times 10^6\ BTU/hr) = (360\ lbm/min)\left(\frac{.24\ BTU}{lbm\ F^\circ}\right)(\Delta T)$$

$$\Delta T = 157\ F^\circ$$

The fuel savings potential associated with pre-heating compressed air is based on adding 819,000 BTU/hr to the air by heat exchanger. In terms of cubic feet of natural gas saved:

$$\text{Fuel savings} = (819{,}000\ BTU/hr)(ft^3/1080\ BTU)/.72$$

$$\text{Fuel savings} = 1054\ ft^3/hr$$

On an annual basis, fuel savings will be 6.677 x 10^6 ft^3/year. At a 1977 fuel cost of $1.36 per MCF, associated dollar savings would be $9,121. The greatest problem associated with this particular concept involves the heat exchanger itself. Because the gas fired heaters at the facility are small relative to boiler units where gas-to-air heat recovery is practiced, off-the-shelf equipment for this specific application does not exist. Instead, a heat exchanger must be specifically designed for this application.

Another area of interest within the Ram Air Facility concerns use of the electrical output from various turbine generators. These generators are loaded to test turbine and generator operation under various conditions. Loading is accomplished by directing generator output to resistor banks. Heat dissipated by the loading process is transferred to the environment by small forced draft air cooling systems.

Direct use of the 400 Hz, 208 V, 3 phase output for electrical power supplementation is possible and there are several avenues of approach. The goal, of course, is to produce 60 Hz electrical energy to insure compatibility with existing electrical systems. Electricity is an excellent energy source because it is a form of work that can be transmitted with relative ease.

Using the 400 Hz generator outputs as a supply to a motor generator pair is one means of generating 60 Hz electricity from the 400 HZ machines, but the present level of technology precludes practical application

(Reference 10). One could also transform and rectify generator output to produce direct current electricity for use without conversion, as a source for a DC motor - AC generator pair, or as a supply to a static inverter to produce 60 Hz AC electricity.

Of the techniques mentioned, the use of a 400 Hz motor - 60 Hz generator pair is the most practical and reliable method available for AC applications. There are problems with parallel operation of the 400 Hz machines, but these are relatively routine and pose no serious difficulty.

General Electric, Schenectady, New York, manufactures motor generator sets capable of producing single or multi-phase. 60 Hz electricity. Additional devices can include optional features to allow for synchronization and parallel operation with an external frequency reference. The per unit cost associated with motor-generation varies according to desired output.

DC electricity may be desired (instead of AC) for charging a battery bank. This approach permits energy storage regardless of the intermittent nature of the supply. General Electric manufactures a transformer rectifier that can produce 28 VDC at 100 amperes (2.8 KW) from a 400 Hz, 208 V, 3 phase source. The cost for this unit is $4,235. A 28 VDC potential is sufficient for charging an array of batteries with nominal operating voltages of 24 VDC, which is the operating voltage for many industrial lift type machines. Charging controls would be necessary to maintain desired charging rates to avoid damaging batteries.

As batteries approach full charge, the amount of current they demand decreases, falling to zero at full charge. To get maximum use from the transformer rectifier units, a continuous supply of batteries calling for high charging rates must be available. This does not appear likely.

The major problem with using the output of the 400 Hz generators in the ways discussed lies with the testing procedures. The primary mission of the Ram Air Facility is to test the air turbines and generators, and whatever electricity generated is incidental to the testing process. A quick review of the testing procedures reveals that generator loading is highly variable and of relatively short duration. For example, the 60 KVA unit, production number 94920A (cf. Table 2.1) is load tested under the following KW load/time conditions: 45 KW/5 min; 20 KW/5 min; 45 KW/1 min; 10 Kw/1 min; 4 KW/1 min; 30 KW/1 min; and 4 KW/1 min. The total time this unit is tested is 12.1 hours, yet the total time it is loaded is 25 minutes. This is typical of loading and time relationships at the Ram Air Facility. The fact that energy/time ratios are small must be considered in weighing costs and benefits.

Any use of the electrical output from the units under test must be compatible with the testing procedures, particularly with regard to loading flexibility. If a test calls for a 10 KW load to be applied, the requirement must be met. The same test may later call for a 45 KW load, and this requirement must also be met, quickly and easily. Motor-generator sets or transformer rectifiers do not allow for flexibility in loading directly in that the load they impose on a turbo-generator

is directly proportional to the load placed on them, and this may be difficult to control.

A less efficient (from an availability standpoint) alternative for using generator output centers around incorporation of the electric output as a supply to electric air heaters for use in preheating air used for the air turbines. The preheating would take place before air heating by the gas fired furnaces installed for this purpose. The gain realizable is in terms of reduced heating requirements for the gas furnaces, and therefore, decreased natural gas demand.

Table 12.1 serves to illustrate the typical type of equipment under test at the facility. Assuming each of the generators listed are undergoing testing simultaneously, the total actual input to the compressed air via electric air heaters is 43,674 BTU/hr. In terms of cubic feet, this represents 53.92 cubic feet of natural gas saved per hour by utilization of electric air heaters supplied by test generator output. As shown, the natural gas saving is 53.92 ft^3/hr or 1294.08 ft^3/day. At gas rates of $1.366 per MCF, the yearly (260 days) dollar saving realizable is $459.61. At the projected 1982 rates, yearly savings will escalate to $993.90 and by 1977 (7% increase in rates per year from 1982 to 1997) will be $2647.00 per year per compressor in operation, (12.80 KW total continuous) based on $7.867 per MCF cost for natural gas by that year.

These calculations are based on a 43,674 BTU/hr KW continuous input to the compressed air from the air heaters. Units under test are not operated at full load on a continuous basis, but there is usually

Table 12.1. Representative air turbine test data.

Production Number	Rating (400 cps)	Total BTU per test	Total time for test	BTU/hr Potential	BTU/hr with 83% air heater efficiency	Heater Type	Unit Cost
31174A	20 KVA, 3Ø 120/208 volts	35,372.9	4.9 hours	8,843.23	7,339.88	GCH 62005	971.00
93031A	20 KVA, 3Ø 120/208 volts	35,372.9	4.9 hours	8,843.23	7,339.88	GCH 62005	971.00
30151A	60 KVA, 3Ø 120/208 volts	131,506.95	13.16 hours	9,992.93	8,294.13	GCHB 850015	2027.00
93005A	12 KVA, 3Ø 120/208 volts	54,608.00	4.59 hours	11,897.17	9,874.65	GCH 61205	796.00
93022	12 KW, 400 Amp 30 VDC	42,776.27	5.23 hours	8,179.02	6,788.59	N/A	N/A
94920A	60 KVA, 3Ø 120/208 volts	31,172.07	12.1 hours	2,576.2	2,138.25	GCHB 850015	2027.00
93507	60 KVA, 3Ø 120/208 volts	69,539.88	9.35 hours	7,437.42	6,173.06	GCHB 850015	2027.00
93012	3.5 KVA, 3Ø 120/208 volts	3,833.94	1.52 hours	2,522.33	2,093.53	GCH 3405	397.00
93010	5 KVA, 3Ø 120/208 volts	5,688.33	11.22 hours	506.98	420.79	GCH 3605	500.00

more than one unit in operation at any given time, (up to 12 max.) so that the total electrical output for the facility could be above the 12.80 KW level. Therefore, a continuous average of 12.80 KW additional heat to the air stream is not deemed unreasonable.

Because of difficulties involved with parallel operation of the generator being tested, it is not economically or practically feasible to simultaneously supply a single air heating device with multiple sources. The implication, of course, is that each air heating unit must be supplied by a single generator at any given time. Total initial expenditure for the air heating units must therefore be adjusted according to the number of units installed and the unit cost, based on KW rating of the unit. If the eight air heating units proposed were installed to accommodate typical units tested, the initial cost outlay (units only) would be $10,000. Figure 12.7 illustrates the savings with the air heating units in operation (at 12.80 KW continuous) heating 80 lbm/min air flow (one compressor). Assuming again that the total electrical input to the air heaters is 12.80 KW continuous, the break-even point as shown in Figure 12.8 would occur in the 14th year after installation according to present sum analysis. Note that the break-even point specified includes total unit cost only and does not specify auxiliary and installation costs. It appears that utilization of resistance heaters is not justifiable at this time.

Clearly, the savings realizable by incorporation of air heating devices supplied by test generator output are in direct proportion to

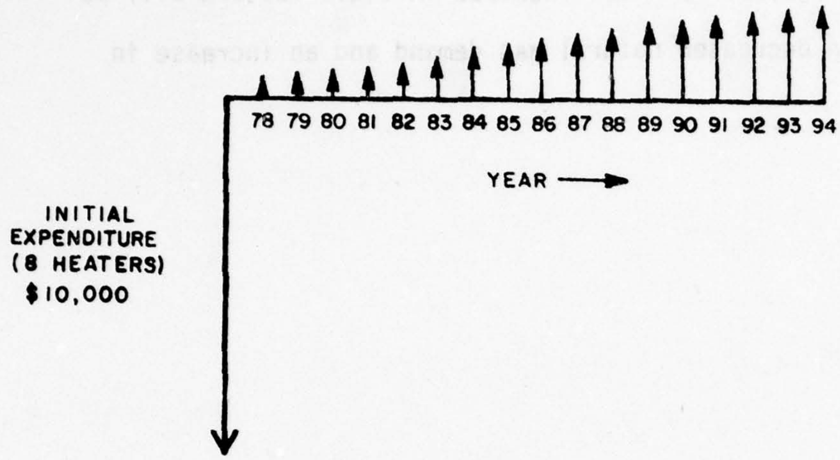


Figure 12.7. Savings schedule for electric air heaters at the Ram Air Facility.

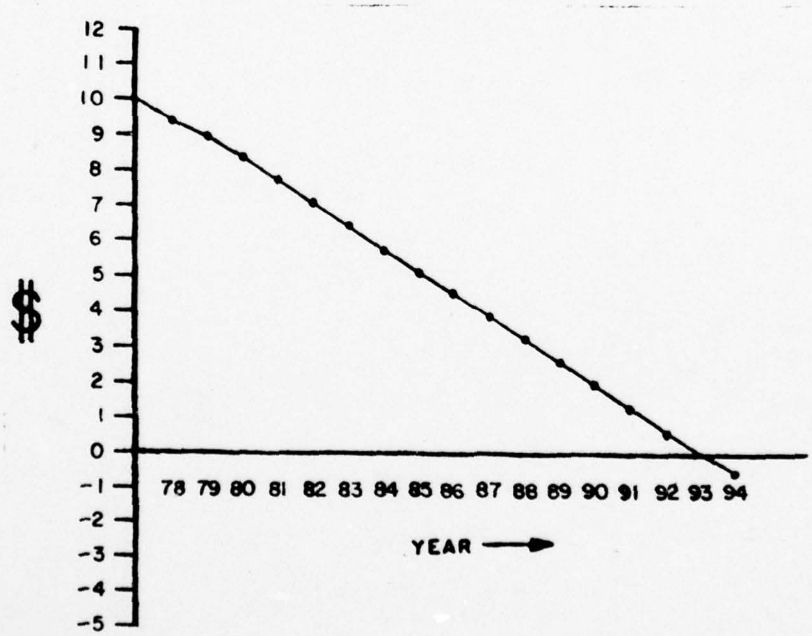


Figure 12.8. Present sum analysis for electric air heaters at the Ram Air Facility.

the amount of load placed on the generators and the amount of time they are in operation at a given load. An increase in these factors will be directly reflected in decreased natural gas demand and an increase in dollar savings.

12.3. Optimization of Energy Utilization at the Ram Air Test Facility

In many cases, the recovery of rejected energy can be significantly improved if overall aspects of a system are considered. In this case, the energy efficiency of the system can be improved if the energy required for compression and energy required for air heating can be combined into a single system. An additional advantage is gained because the new system can use waste fuel.

Waste petroleum, oil and lubricants (POL) pose a disposal problem for the Air Force, and combustion in gas turbines or boilers offers solutions. Gas turbines offer a direct means of extracting shaft work from the expansion of hot gases produced from burning of fuel. They exhibit a wide range of fuel flexibility and gas/liquid fuel switching can be accomplished without interruption of turbine operation. Shaft work can be used in a variety of ways including electrical generation and mechanical work. Gas turbines are well suited as prime movers for reciprocating machines such as air compressors. In addition to usefulness as prime movers, gas turbines used in conjunction with a combined energy system may provide heat in much the same way as a conventional boiler. A combined energy system is designed to extract thermal energy from hot (900°F) exhaust gases exiting the gas turbine.

Installing a pilot gas turbine unit at the Ram Air Test Facility as a replacement for a single 800 HP electric motor currently being

used as a prime mover for a three-stage reciprocating air compressor could improve the overall efficiency of the facility. The advantages would be: (1) Elimination of an 800 HP electric motor, (2) Ability to extract useful work from waste JP-4 and JP-5 (2,400 gal/month at Tinker AFB) and (3) Ability to burn natural gas for work, using exhaust for air heating.

The basic gas turbine process involves a compressor, burner, and expander. Air is usually inducted into the compressor at atmospheric pressure and water or alcohol may be injected into the air stream to increase the volumetric efficiency of the compressor by cooling. The combustion process takes place in the burner section with temperature limitations imposed by turbine blade materials and cooling requirements. A portion of the air from the compressor (primary air) is mixed with fuel and combustion is initiated. The remaining air (secondary air) is mixed with the hot gas to moderate temperature for optimum turbine performance. The combustion products and secondary air ideally undergo an adiabatic expansion process in the turbine section. The turbine section is mechanically coupled to the compressor as a work input. Shaft work is also extracted from the turbine directly or through reduction gears, depending on the application.

The Saturn Gas Turbine manufactured by Solar Turbines International of San Diego, California is an 1100 HP unit which would be ideal for the proposed alteration at the Ram Air Test Facility. This unit is a two-shaft variable speed engine weighing 8,250 pounds. It requires a fuel supply, a gas or air starter system, and a 110 VAC 60 HZ power source.

Some basic calculations concerning gas turbine operating parameters are done to aid in determining advantages associated with turbine use. With an inlet temperature of 59^oF and zero pressure losses in the inlet and exhaust ducting, the unit horsepower rating is 1150 HP with a specific fuel consumption of 11,500 BTU/HP/hr. Horsepower rating and fuel consumption are affected by altitude, inlet temperature, and inlet/outlet pressure losses. Assuming pressure losses for the inlet and outlet piping of 2 inches H_2O, 70^oF turbine inlet temperature and an altitude of 2300 feet, the actual horsepower rating will be 1014 HP. The power demand on the unit will be 800 HP determined by the reciprocating compressor. The exhaust temperature and mass flow rate are determined using the 800 HP restriction and are 775^oF at 11.4 lbm/sec. The adjusted specific fuel consumption is 12,366 BTU/HP/hr. This translates to 9.89×10^6 BTU/hr at an 800 HP demand load. In terms of natural gas, 9160 cubic feet will be combusted per hour, while waste JP-5 will be consumed at a rate of 540.59 lbm/hr (83.5 gal/hr). Waste JP-5 at Tinker AFB is accumulated at a rate of about 2,400 gallons per month, and therefore the turbine unit could be operated on waste fuel for 29 hours per month or about 6% of the normal facility operating time.

Clearly, burning waste fuels alone cannot justify the installation of a gas turbine unit at the Ram Air Facility because the fuel source is limited to 2400 gallons per month. The gas turbine unit is capable of operating with a dual fuel supply so that it will be possible to operate the unit on natural gas as well as JP-5. The combined energy package allows the turbine exhaust gases to be used for heating the compressed air that is normally heated by the two large natural gas fired furnaces at the

Ram Air Facility (RAF). For this reason, natural gas can be diverted from one furnace for use as gas turbine unit (GTU) supply.

As stated earlier, the GTU exhaust parameters at 800 Hp are 775°F at 11.4 lbm/sec. The amount of thermal energy being rejected from the turbine is:

$$Q_{out} = 11.4 \frac{\text{lbm}}{\text{sec}} \left(\frac{.245 \text{ BTU}}{\text{lbm F}^{\circ}}\right) (775-70)$$

$$Q_{out} = 1969.07 \text{ BTU/sec} = 7.09 \times 10^{6} \text{ BTU/hr}$$

Each gas fired air heater is designed to heat 21,600 lbm/hr air from 75°F to 1000°F. In so doing, each furnace uses 6770 cubic feet of natural gas. The design energy flow for a furnace is:

$$Q_{in} = 6 \text{ lbm/sec} \left(\frac{.240 \text{ BTU}}{\text{lbm F}^{\circ}}\right)(1000-75)$$

$$Q_{in} = 1332 \text{ BTU/sec} = 4.79 \times 10^{6} \text{ BTU/hr}$$

The amount of thermal energy rejected by a GTU is greater than that imparted to the compressed air by a single furnace. The gas consumption rate is higher for the GTU than for the furnace by 2390 cubic feet per hour. This is equivalent to 2129 cubic feet of natural gas while the actual difference in the amount of gas used is 2390 cubic feet per hour. In addition, shaft work is being extracted from the GTU at a rate of 800 HP.

Not all of the heat rejected from the GTU can be transferred to the air because of heat exchanger efficiency. Using the representative figure of 75%, the heat that can be transferred to the air is 5,317,500 BTU/hr. The air temperature attainable using the combined energy system will be approximately 725°F at the design compressed air flow rate of 6 lbm/sec; the equivalent output of 4 1/2 compressors. Additional heating will be necessary to meet transient 750°F air heating requirements, and this can be accomplished with the second existing gas fired air heater.

The economic analysis of the proposed modifications at the RAF begins with the comparison of the savings associated with installation of a GTU, and weigh these against the capital outlay. The benefits or savings are: (1) Elimination of a 596 KW (800 HP) electric motor load; (2) Reduced compressed air heating requirement by the natural gas fired furnace; (3) Productive utilization of a "waste" fuel source. The assumptions made for the savings calculations are: (1) 800 HP load imposed on the GTU whenever it is in operation; (2) Continuous source (2400 gallons/month) of waste JP-5.

The first contribution to the savings will be from a reduced electrical demand of 596 KW, 24 hours per day, 260 days per year. The annual escalation rate for electricity is 16% for 1977-1980, and 10% per year thereafter. The savings schedule is shown in Table 12.2.

YEAR	SAVINGS ($)
1978	86,437
1979	100,267
1980	116,310
1981	131,172
1982	144,289
1983	158,718
1984	174,590
1985	192,049
1986	211,254
1987	232,380
1988	255,618
1989	281,179
1990	309,297

Table 12.2 Savings schedule for reduced electrical demand by utilization of the GTU at the Ram Air Facility.

The second aspect of possible savings will be the reduced air heating requirement for the natural gas fired furnaces. Because there are usually three compressors in operation at any given time, the air heating requirements and subsequent natural gas calculations will be based on the air flow from three compressors. Each compressor provides 200 lbm/min air flow, but only 40% (80 lbm/min) is required to be heated; therefore, the total heated air flow from three compressors will be 240 lbm/min. The air is heated from $50^{o}F$ to $1000^{o}F$. and the natural gas consumption for a 75% efficient furnace is calculated as follows.

$$Q_{gas} = 240 \text{ lbm/min} \times .24 \text{ BTU/lbm-F} \times 950^{o} \text{ F.} \times 1.33$$

$$Q_{gas} = 4.367 \times 10^{6} \text{ BTU/hr}$$

$$FT^{3}_{gas} = 4,043 \text{ cubic feet per hour}$$

The amount of natural gas used by the GTU at an 800 HP load is 9160 cubic feet per hour or more than twice the amount used by a single furnace. However, the combined system will permit GTU exhaust gases to be used for compressed air heating and is sufficient to heat the total airflow from all the compressors (6) at the RAF to 725°F. Additional heating will be required to meet transient demands and the amount of gas needed for this purpose is:

$$Q_{in} = 240 \text{ lbm/min } \{\frac{.24 \text{ BTU}}{\text{lbm F}^o}\} \{275^o \text{ F.}\} \times 1.33$$

$$Q_{in} = 1.264 \times 10^6 \text{ BTU/hr}$$

$$Ft^3{}_{gas} = 1.264 \times 10^6 \text{ BTU/hr} \times \frac{1 \text{ Ft}^3}{1080 \text{ BTU}}$$

$$Ft^3{}_{gas} = 1{,}170 \text{ cubic feet per hour}$$

The total gas requirement for GTU operation will be 10,330 cubic feet per hour as opposed to the current demand of 4,043 cubic feet per hour or 2.134×10^6 Ft^3 per month (24 hrs/day, 22 days/month). The monthly gas demand for the GTU must be adjusted to consider the contribution from waste JP-5, therefore, the monthly natural gas demand for the GTU is based on 24 hrs/day, 20.79 days/month. With this adjustment, the GTU monthly demand is 5.15×10^6 ft^3/month. This is 3.02×10^6 cubic feet per month more than is currently being used. Cost analysis results are shown in Table 12.3 and illustrate the added cost of using the GTU at the RAF.

YEAR	COST
1978	57,990
1979	66,688
1980	73,838
1981	79,006
1982	84,537
1983	90,455
1984	96,786
1985	103,561
1986	110,811
1987	118,568
1988	126,867
1989	135,748
1990	145,250

Table 12.3. Yearly natural gas cost increase incurred by using a gas turbine unit at the Ram Air Facility.

There are no natural gas savings. In fact, the net overall demand increases by 36 million cubic feet per year. However, a comparison between Table 12.2 and Table 12.3 illustrates quite clearly that this added cost is more than offset by the electrical cost reduction. The 12 year analysis indicates that the savings in electricity cost are greater than the added gas cost by $1,093,455 or roughly $92,000 per year.

The annual savings illustrated in Table 12.4 represent the net savings determined from electric use reduction, natural gas use increase, and loss of income from waste JP-5. (A private contractor currently pays 16.1 cents per gallon for waste JP-5, amounting to $4,637 per year).

The costs associated with the installation of a gas turbine unit are $220,000 to $275,000. This is an installed price for the turbine unit and ancillary equipment. An additional cost of $22,000 will include the energy recovery package, bringing the total installed cost to $297,000. In addition, yearly maintenance costs are estimated to be $5,000.

YEAR	NET SAVINGS ($)
78	23,810
79	28,942
80	37,835
81	47,529
82	55,115
83	63,626
84	73,167
85	83,851
86	95,806
87	109,175
88	124,114
89	140,794
90	159,410

Table 12.4. Net Annual Savings

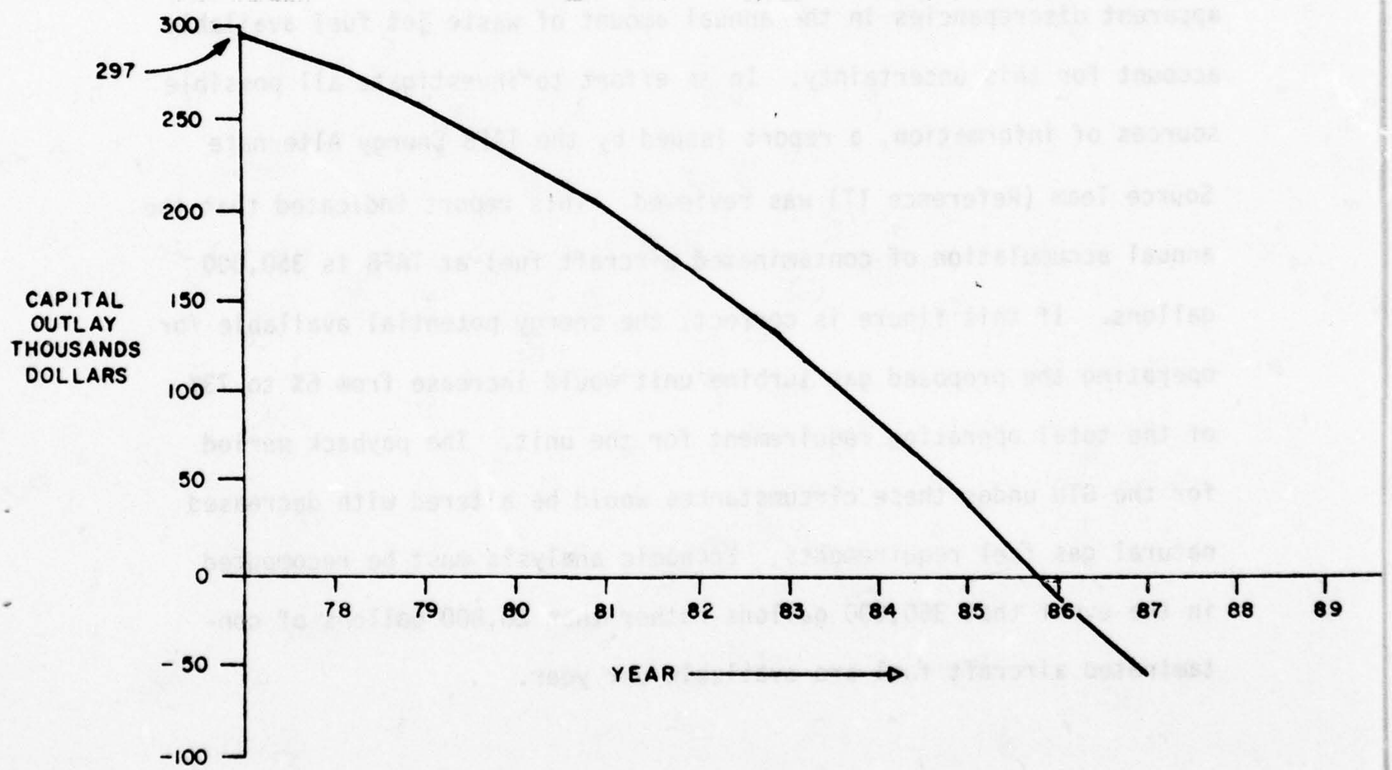


Figure 12.9. Present sum analysis for gas turbine unit.

Figure 12.9 illustrates the results of a present sum analysis based on an initial outlay of $297,000 net annual savings from Table 12.4 and yearly maintenance costs of $5,000.

The present sum analysis indicates a payback period of 8.6 years. It is interesting to note that the payback period is as long as it is due to the relatively small initial annual net savings. As Table 12.4 indicates, continued escalation of savings occurs, due to increased savings in electrical use overriding increasing natural gas costs.

It is possible that the payback period may be somewhat different; apparent discrepancies in the annual amount of waste jet fuel available account for this uncertainty. In an effort to investigate all possible sources of information, a report issued by the TAFB Energy Alternate Source Team (Reference 11) was reviewed. This report indicated that the annual accumulation of contaminated aircraft fuel at TAFB is 350,000 gallons. If this figure is correct, the energy potential available for operating the proposed gas turbine unit would increase from 6% to 73% of the total operating requirement for the unit. The payback period for the GTU under these circumstances would be altered with decreased natural gas fuel requirements. Economic analysis must be recomputed in the event that 350,000 gallons rather than 28,800 gallons of contaminated aircraft fuel are available per year.

Recall that the amount of jet fuel burned by the GTU at 800 HP is 83.5 gallons per hour. With this fuel usage rate, 350,000 gallons of fuel would suffice to operate the GTU for 4,192 hours per year. Assuming a 24 hour per day, 264 day work year, 66% of the GTU fuel needs could be supplied by this waste fuel source, reducing the need for natural gas accordingly. Table 12.5 illustrates the gas costs associated with 34% of the GTU fuel needs supplied by natural gas rather than 94% as in Table 12.3.

YEAR	COST ($)
1978	2,183
1979	2,511
1980	2,887
1981	3,197
1982	3,421
1983	3,660
1984	3,916
1985	4,190
1986	4,484
1987	4,793
1988	5,133
1989	5,492
1990	5,877

Table 12.5. Yearly natural gas cost increase incurred using a gas turbine unit at the Ram Air Facility.

Evidently, gas costs are reduced considerably by using waste or contaminated aircraft fuel. Net savings are calculated and are the result of electric savings, minus gas cost, minus income from the sale of contaminated fuel (@ 16.1¢ per gallon). The savings schedule is shown in Table 12.6. The price paid for the fuel by a contractor will be allowed to increase at 7% per year.

YEAR	SAVINGS ($)
1978	25,804
1979	35,456
1980	46,573
1981	56,575
1982	64,568
1983	73,158
1984	83,174
1985	94,059
1986	106,674
1987	120,482
1988	135,685
1989	152,837
1990	172,170

Table 12.9. Net annual savings at RAF using 350,000 gallons of contaminated fuel per year for GTU operations.

A comparison of the results from this analysis with those represented in Table 12.6 reveals that net savings realizable using 350,000 gallons of contaminated fuel are not much greater than for 28,800 gallons. The great reduction in the amount of gas used is offset by the loss of income from the sale of the contaminated fuel. A present sum analysis is illustrated in Figure 12.10 and represents a $297,000 initial outlay, $5,000 annual maintenance cost, and savings from the savings schedule shown in Table 12.6. The payback time using the 350,000 gallon availability for waste fuel is 7.75 years or about one year shorter than for 28,800 gallons.

Gas turbine units are reliable and relatively clean burning, and the combined energy system permits overall efficiencies in excess of 70%. The GTU as applied to the Ram Air Facility would eliminate a large electrical demand, and provide a means for beneficial disposal of waste fuel. The exhaust from the proposed single unit would be sufficient to meet a large percentage of the heating requirement for the compressed air system, with supplemental air heating provided by installed gas furnaces.

Economic analysis provides additional incentive for installation of a pilot gas turbine unit at the facility, with net annual savings rising to $159,000 after 12 years. All aspects considered, the gas turbine holds promise toward meeting the Air Force goal of energy efficiency.

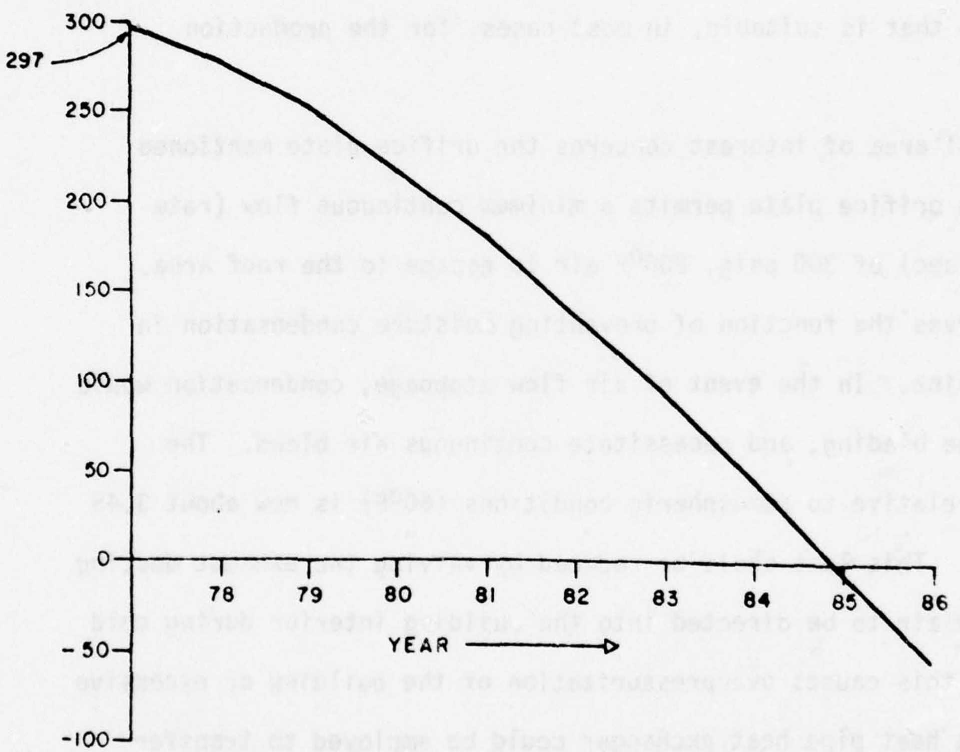


Figure 12.10 Present sum analysis for GTU operating on 350,000 gallons of contaminated aircraft fuel.

12.4. Utilization of Exhaust Compressed Air

At many compressed air facilities, excess air is often exhausted to maintain clean feed lines or manifolds. This is the case at the Ram Air Test Facility. Although the results here apply to this specific facility, they do present a possible use for a rejected energy source that is suitable, in most cases, for the production of work.

The final area of interest concerns the orifice plate mentioned earlier. The orifice plate permits a minimum continuous flow (rate of 0.541 lbm/sec) of 300 psig, $800^{o}F$ air to escape to the roof area. The plate serves the function of preventing moisture condensation in the hot air line. In the event of air flow stoppage, condensation would damage turbine blading, and necessitate continuous air bleed. The energy loss relative to atmospheric conditions ($60^{o}F$) is now about 3.45×10^{5} BTU/hr. This loss could be reduced by valving the exhaust ducting to enable the air to be directed into the building interior during cold weather. If this causes overpressurization of the building or excessive noise, then a heat pipe heat exchanger could be employed to transfer heat from the exhaust air to building air or to run an absorption air conditioner if conditions justify.

The rejected energy source is, however, at high pressure and therefore directly able to produce work. An air turbine for power

generation could be employed, and potentially, one could generate 2.57×10^5 BTU/hr of electrical power with a 75% efficient turbine generator system exhausting at atmospheric pressure. Since this would be operating continuously for five days per week, one could generate 900 KW-hr/week of electrical power which could be pumped into existing power lines.

Perhaps the simplest means of generating electrical energy would be to use an appropriate number of air driven turbine generator units (75 KW output total) which have been rejected for aircraft use but are still capable of producing power. One possible use for this energy, although perhaps not the best possible use, is to supply air in-line electric air heaters placed just up-stream of the gas fired air heaters. In this application, no conversion synchronizing equipment would be needed, as would be the case for its direct use of the electricity. The turbine generator unit could simply supply the electric air heater directly, at 208 volts, 400 HZ. Assuming a heater efficiency of 95%, the amount of energy that such an arrangement could add to the compressed air is then 2.44×10^5 BTU/hr. Given a compressed air mass flow rate of 12 lbm/sec (maximum for two gas fired heaters), and an air inlet temperature of 75^{o}F, the addition of 244,000 BTU/hr could pre-heat the compressed air to 99^{o}F. The natural gas savings would amount to 315 ft^3/hr or 9,960,000 ft^3 per year. Assuming a price of $1.366 per MCF for natural gas, the savings in dollars represented by implementation of this scheme would be $2700. An electric heater that would meet this application is the

Chromalux model GCHB 850015, costing $2027. It is evident this will quickly be repaid, assuming surplus turbine generator units can be used.

12.5. Possible Resulting Recovery Scheme for the Ram Air Test Facility

The recovery of rejected energy at the Ram Air Facility involves use of several sources including: 1) cooling water effluent, 2) high temperature compressor discharge, 3) combustion exhaust, 4) hot air vent orifice. Recuperative heat exchange between cooling water effluent and air exiting the chiller unit promises significant energy savings. Use of high temperature compressed air exiting the air compressors can provide building heating as required. Preheating compressed air before entry into the gas-fired air heaters by gas-to-air heat exchange with heater exhaust, can assist in reducing natural gas oonsumption. Installation of an air driven turbine generator placed in the hot air vent orifice air stream, coupled with an instream electric air heater, can further reduce gas consumption. Replacement of a single 800 HP electric motor serving as the prime mover for a reciprocating air compressor with a gas turbine unit could provide significant electrical savings. Another benefit associated with a gas turbine involves waste POL utilization. Replacement of the Rankine cycle air chiller unit with a desiccant filtration system will reduce electrical and natural gas demand. Because all the proposed recovery equipment cannot be used simultaneously, two modification schemes are proposed. Figure 12.11 shows one scheme involving the gas turbine unit, while Figure 12.12 shows the recuperative and electric air heating units.

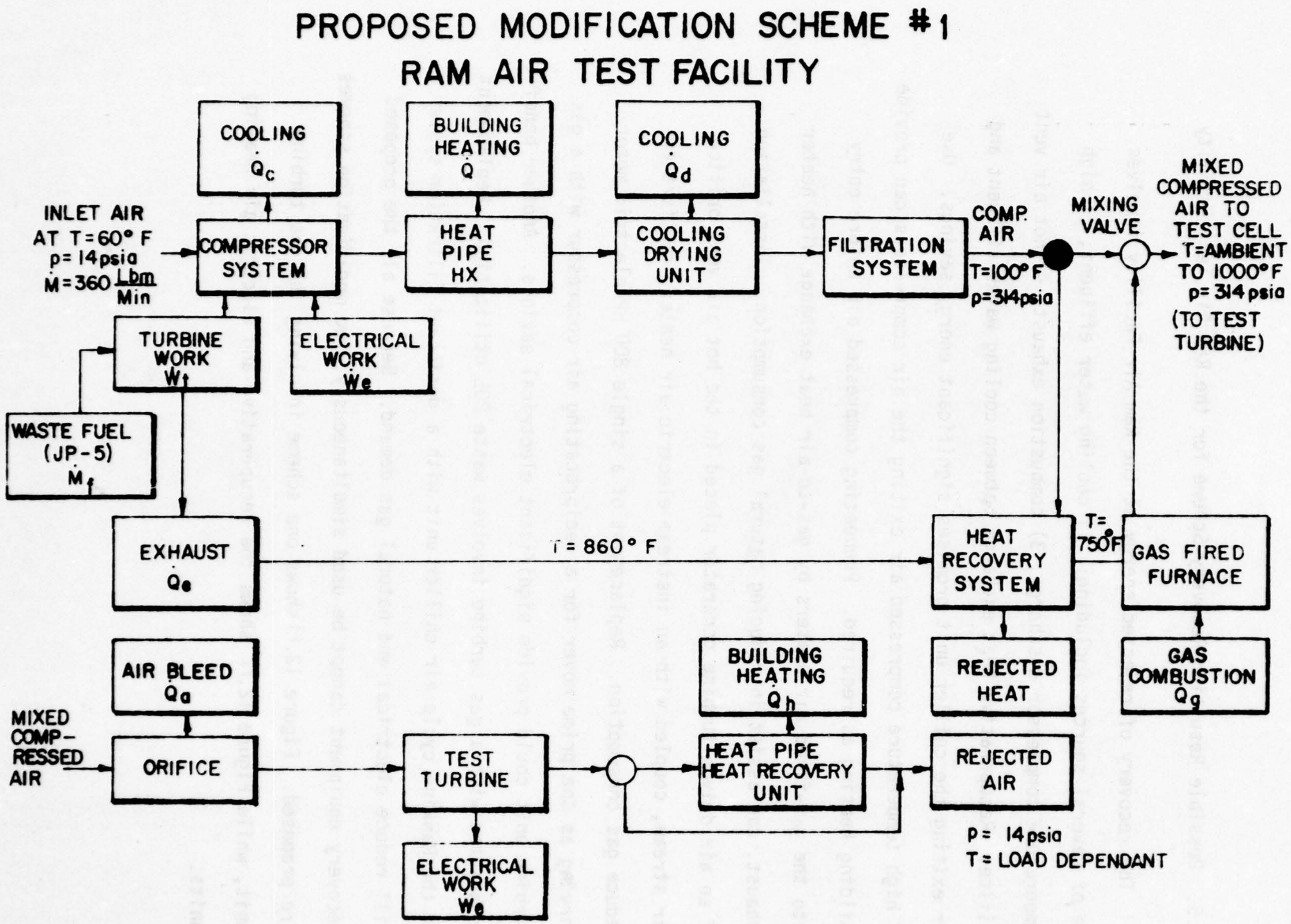


Figure 12.11 Proposed modification scheme #1 for Ram Air Test Facility

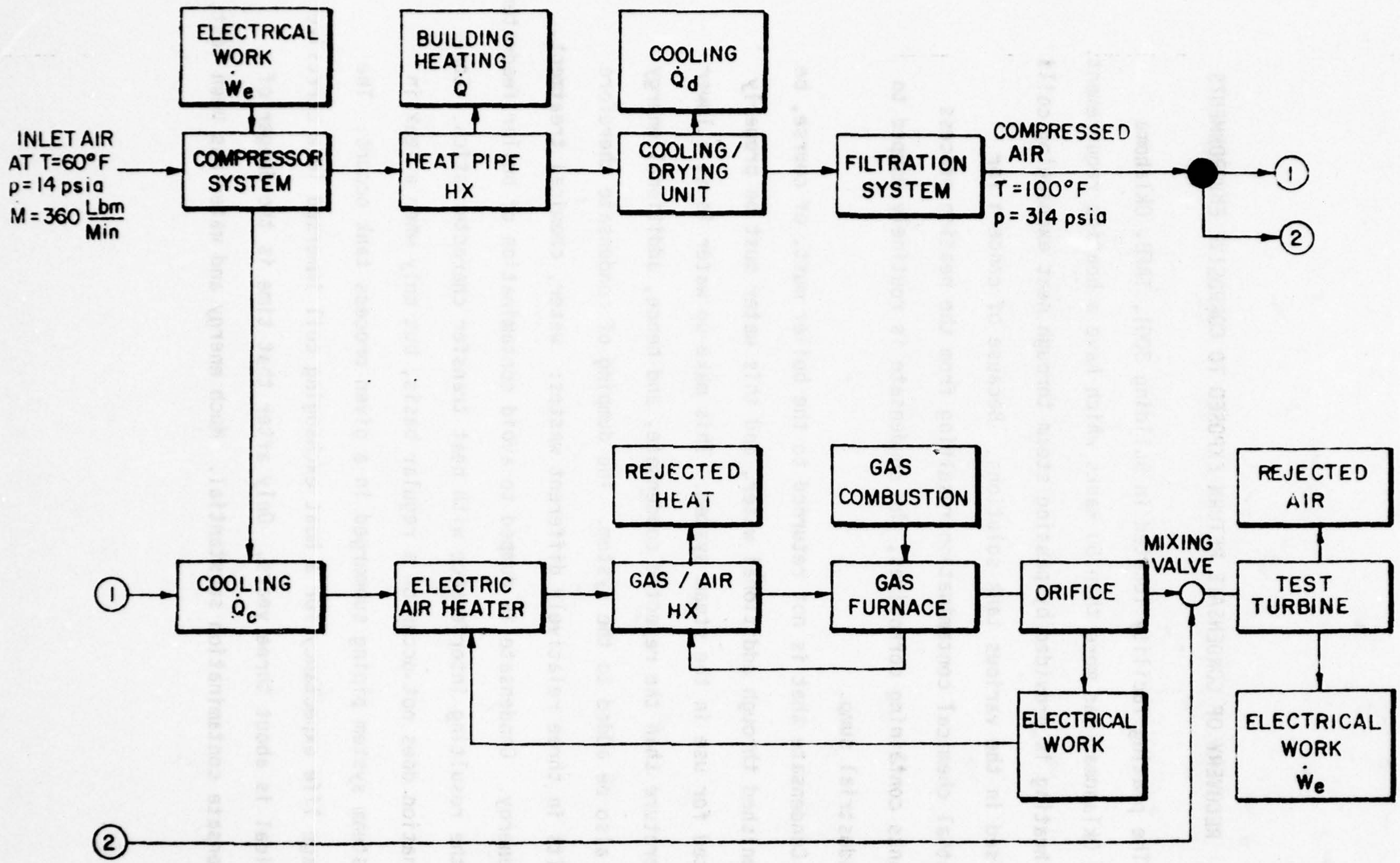


Figure 12.12 Proposed modification scheme #2 for Ram Air Test Facility

13.0. RECOVERY OF CONDENSATE RETURN EXPOSED TO CORROSIVE ENVIRONMENTS

The plating facility located in Building 3001, TAFB, Oklahoma City, Oklahoma, has more than 80 tanks which have a heating requirement. Tank heating is provided by passing steam through heat exchanging coils immersed in the various tank solutions. Because of concern for potential chemical contamination resulting from the heating process in tanks containing corrosives, the condensate is routinely dumped to an industrial sump.

Condensate that is not returned to the boiler must, of course, be replenished through additional water, and this water must be properly treated for use in the steam system. This make-up water is at a lower temperature than the rejected condensate, and hence, additional energy must also be added to the system. The dumping of condensate therefore results in three relatively different wastes: water, chemical treatment, and energy. Condensate is dumped to avoid contamination of boiler feedwater and the resulting interference with heat transfer characteristics. Contamination does not occur on a regular basis, but only when a leak in the steam system piping submerged in a given process tank occurs. The average life expectancy for a heat exchanging coil immersed in a corrosive chemical is about three years. Only after that time is the danger of condensate contamination substantial. Much energy and water has been lost,

however, in the practice of dumping condensate to avoid a potential problem.

Condensate from all tanks could be returned to the boiler if water quality could be assured. Several devices exist for monitoring various water characteristics, such as PH detectors and conductivity cells. If a change in parameters occured indicating a heat exchanger leak, condensate could then be dumped to the drain, and appropriate alarms and indications could be activated. An isolation procedure could be used to determine the location of the faulty heat exchanger. Such a system has the additional benefit of rapid indication of heat exchanger failure. The installation of this type of detection equipment would require periodic maintenance, with the routines consisting primarily of cleaning probe surfaces, which is usually done quickly and easily.

To quantify the potential savings from the addition of such equipment, one must know the flow rate and temperature of the condensate as it is dumped. For the TAFB plating facility, five gallons per minute were dumped at 190°F. The amount of energy lost compared to the required additional energy to make up water at 50°F is:

$$Q = (2500 \frac{\text{lbm}}{\text{hr}})(\frac{1 \text{ BTU}}{\text{lbm F}^{\circ}})(190^{\circ}\text{F}-50^{\circ}\text{F})$$

$$Q = 350{,}000 \text{ BTU/hr}$$

Allowing for an 85% loss in the condensate return line:

$$Q = 300,000 \text{ BTU/hr}$$

This energy is lost on a continuous basis and amounts to 1.8×10^9 BTUs per year based on 24 hours per day, 254 days per year of operation. Assuming the boiler is 80% efficient, the amount of natural gas that must be consumed by the boiler to replace that lost through condensate dumping is:

$$\text{ft}^3 \text{ Nat. gas} = \frac{1.8 \times 10^9 \text{ BTU}}{(.8)(1080 \text{ BTU/ft}^3)}$$

$$\text{ft}^3 \text{ Nat. gas} = 2.1 \times 10^6 \text{ per year}$$

At the current (1977) Tinker AFB gas rate of $1.366 per MCF, the cost of dumping condensate is $2900.

Escalating natural gas cost savings at 15% per year until 1980, and 8% thereafter, result in the cost saving schedule illustrated in Table 13.1.

Year	Fuel Cost Savings
1978	$3,344
1979	3,846
1980	4,423
1981	4,777
1982	5,159
1983	5,572
1984	6,017
1985	6,499
1986	7,019

Table 13.1. Savings schedule for condensate return

Incorporating condensate return in this instance would be both practical and beneficial. The monitoring device used for analytical purposes is the Beckman PH monitor, Model 941. This unit is capable of operating at temperatures up to 100^{o}C, and can open a dump valve for both high and low PH, as well as operate an alarm system. The associated initial capital outlay for the PH monitor, additional valving and the control and alarm system is estimated to be $1500. Assuming an installation cost of $750 and a maintenance cost of $4 per week, the break-even time is quite rapid and is shown graphically in Figure 13.1. This represents only the energy savings of Table 13.1 and not water or chemical treatment savings. Even if such a return line were only operable 50% of the time, it would pay for itself in less than two years.

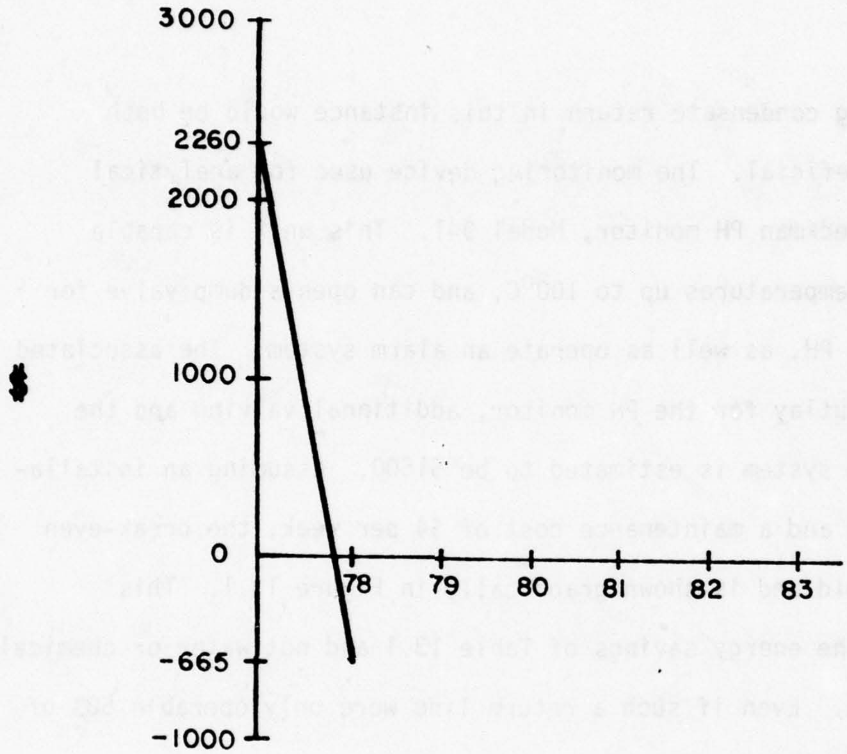


Figure 13.1 **Present sum analysis for condensate return at the plating facility, TAFB.**

14. RECOVERY OF REJECTED ENERGY FROM A JET ENGINE TEST CELL

14.1 General Considerations

The basic goal of the Air Force jet engine test cell facility is to certify engines for military use. In the process of certification, certain basic parameters germane to the engine performance are measured such as thrust, flow rates, exhaust gas temperature, exhaust pressure, et cetera. Nearly all the power generated by the combustion process is given up in the exhaust, but no work is produced because the engine remains virtually motionless on the test stand. In a given test, the power can be quite large, as an engine such as the TF33 (used on the B-52H and C-141) will burn fuel at the rate of 13,000 lbs/hr at full power (no afterburner). This represents 2.4 x 10^8 BTU/hr or 70 megawatts of combustion energy. The majority of engine testing is typically done at 75% to 100% power levels.

Currently, a jet engine test cell is equipped with an augmentor tube and exhaust stack to dissipate the power exhausted by the engine. The augmentor tube dissipates the exhaust kinetic energy and lowers the sound level in the exhaust. The jet engine exhausts directly into the constant area augmentor tube. The tube is not connected to the jet engine, and a certain amount of ambient air is entrained by the exhaust, depending upon the engine size and the power level at which it is being tested. The entrained air not only dissipates exhaust kinetic energy, it also contributes to lowering the temperature of the exhaust gases.

As power level is increased, and hence the mass flow rate through the engine increased, less ambient air is entrained. Water is then sprayed in the augmentor tube to lower the temperature of the exhaust gases. This typically occurs when the augmentor tube temperatures exceed 350°F. The water spray is sufficient to keep the augmentor tube temperatures near this value. The amount of water sprayed varies, although when engines are tested in the afterburner mode, as many as 400 gal/min of water may be used to moderate the engine exhaust.

The after section of the augmentor tube is perforated and contained within the exhaust stack and is called the diffuser. The total area of the perforations are no less than the intake area, and the end of the tube is closed. Here, the exhaust kinetic energy is fully dissipated and the exhaust subsequently passes out the exhaust stack. The exhaust stacks are perforated to dampen the high noise level of the exhaust. Much of the water spray that has been evaporated in the augmentor tube condenses here.

Augmentor tubes and exhaust stacks were designed to dissipate energy, with no consideration given to the recovery of any of the exhaust energy. Fortunately, the augmentor tube is not built into the building and could be replaced if necessary. Besides the intermittent nature of this rather large power waste energy source (a problem which plagues most waste energy sources of high quality), other major problems exist which make optimum recovery of the exhaust energy difficult. The severest of these is the fact that the augmentor tube must be decoupled from the jet engine for suitable jet engine testing.

The limiting critical parameter in jet engine performance is the turbine inlet temperature. Since engine performance is extremely sensitive to this parameter, every attempt is made to make it as high as possible. Although the adiabatic flame temperature of jet fuel is on the order of $4000^{o}R$ (2222^{o} K), turbine inlet temperatures are limited to 2000^{o} R (1111^{o} K) because of the materials limit of the turbine blades.* Nearly all engines operating today operate at this limit so that performance may be maximized. Due to this fact, the exhaust gas temperature (EGT) of virtually all engines are nearly equal. The EGT is measured just downstream of the turbine exit. The turbine will extract sufficient energy to drive the compressor, and in the case of turbofans, an additional component of energy to drive the fan. The net result is an EGT in the range of 1500^{o} R (833^{o} K) to 1700^{o} R (944^{o} K). The EGT is a stagnation temperature, since it is measured in the region where the flow velocity is relatively low. The exhaust nozzle transforms a portion of this energy into kinetic energy. Most engines being rebuilt and tested today are designed so that the exhaust nozzle operates with a local Mach number near one. The exhaust velocities, however, are about 1500 to 1900 ft/sec. These high velocities generate severe sheer layers with the entrained ambient air. The subsequent entrainment and mixing of the exhaust jet with ambient air result in the dissipation of energy and the loss of mechanical energy.

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*If the first few stages of the turbine blades are cooled, then turbine inlet temperatures of $2500^{o}R$ are possible.

In turbofan engines, the exhaust Mach number is near one but a portion of the air exhausted by the nozzle was compressed by the fan and bypasses the combustor. The downstream mixing lowers the effective EGT and the exhaust velocity. In addition, the initial EGT is slightly lower in a turbofan than in a turbojet, since the turbine must be slightly larger to supply power for compression of the bypass flow in addition to the primary flow. Typical bypass ratios are about 1.5 for the TF30, TF33, and TF41. This means that bypass air mass flow rate is 1.5 times that which passes through the combustor.

When the afterburner is in operation, the EGT is changed substantially while pressure and mass flow rate remain virtually unchanged.* The increase in EGT translates to a substantial increase in exhaust velocity. The EGT with the afterburner in operation will rise to values just less than 4000^{o} R (2222^{o} K), and the exhaust velocity will increase to a value of more than 1.5 times that of its non-afterburning value. Of the engines tested at Tinker AFB, the J-57, J-75 and the TF-30 have afterburners, while the TF-33 and the TF-41 do not. In numbers of engines tested, afterburner engines represent 64% of the total. Although the fuel consumption more than doubles during afterburner firing, afterburners are usually operated only for short periods of time (about 10 minutes), if at all.

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*Exhaust pressure drops slightly and mass flow rate increases slightly.

While EGT represents the most important parameter from the standpoint of energy quality and potential recovery, mass flow rate (MDOT) represents the most important parameter from the standpoint of the rate at which energy can be recovered. Of the engines tested at Tinker AFB, the J-57 is in the 9000 to 17000 Lb_f thrust range; J-57 testing represents about 25% of the engines tested at Tinker AFB. More than half are the 43 model (56% of the 25%), which is used on the B-52G. Since the engines have somewhat different characteristics (turbojet vs. turbofan, afterburning vs. non-afterburning), the values of MDOT span a wide range. Table 14.1 gives approximate MDOTs for the engines tested at Tinker AFB. These values are at full power, and the bypass air is included for the turbofan engines.

Table 14.1. Approximate MDOTs for engines tested at Tinker AFB running at full power.

ENGINE	MDOT (lbm/sec)
J-57	185
J-75	250
TF-30	260
TF-33	450
TF-41	260

The J-75, TF-30, and TF-33 represent 51% of the engines being tested at Tinker AFB, while the J-57 represents 25%, and the TF-41, 24%. These engines span quite a range of MDOTs, and the problem is further complicated by the fact that they are usually operated in a range from 75% to full power.

The considerations of EGT and MDOT, and the fact that the energy is in two different forms make jet engine exhaust relatively unique with regard to waste energy recovery. The approximate distribution of mechanical energy (kinetic energy) and thermal energy (enthalpy) at the exhaust plane is as follows:

$$\frac{\text{Kinetic Energy}}{\text{Total Energy}} = \frac{\frac{V^2}{2}}{C_p\,(T_o - T_a)} = \frac{\frac{\gamma - 1}{2} M^2}{\left(\frac{T_o}{T^*} - \frac{T_a}{T^*}\right)} = \frac{KE}{TE} \qquad (14\text{-}1)$$

Where V is the exhaust velocity, C_p is the specific heat at constant pressure of the exhaust products, T_o is the EGT, T_a is ambient temperature, γ is the ratio of specific heats for the exhaust products, M is the Mach number and T is the local static temperature at the exhaust plane. For jet engines, turbojet and turbofans alike, this ratio is about 20%.

In summary, the jet engines being tested at Tinker AFB display similar exhaust characteristics while also exhibiting a wide range of mass flow rates. The mechanical energy available in the exhaust is about 1/5 of the energy available in the flow.

14.2 Energy Recovery Considerations for a Jet Engine Test Cell

When energy recovery is considered several parameters enter the problem in addition to those previously discussed. These include the ratio of engine operation time to elapsed time and the power level during operation time. Using the Tinker AFB data for engine testing, the yearly average of the ratio of engine operation time to elapsed time can be determined. The projected FY78 production forecast is shown in Table 14.2. The recycle ratio prediction is based upon the recycle ratio experienced over the last five years. Recycle ratio is defined as the number of times that an engine must be tested to meet certification specifications. Although the production forecast calls for 1550 engines, 2558 engines will need to be tested to achieve this goal.

Engine Type	FY78 Production Forecast	Recycle Ratio	FY78 Testing Forecast	% Total
J57	330	1.93	637	25
J75	112	2.51	281	11
TF30	472	1.54	727	28
TF33	218	1.35	294	12
TF41	419	1.48	619	24
TOTAL	1550		2558	

Table 14.2. FY77 engine testing forecast at Tinker AFB.

Operating two shifts five days per week means a total of 4064 hours of jet engine testing. It is estimated that on the average a typical jet engine test takes a total of four hours with the engine running at military power or above. For nearly three hours, therefore, based upon yearly averages, one may expect the following:

$$\frac{\text{Engines Tested}}{\text{Day}} = \frac{2558}{254} = 10 \text{ engines/day}$$

$$\frac{\text{Engine Hours}}{\text{Day}} = 3 \times 10 = 30 \text{ engine hours/day}$$

Because there are two shifts in operation, one may expect that 1.88 engines are operating continuously at the facility. There are twelve test cells at Tinker AFB; the average usage time per cell is therefore 15.7%. This translates to an average of slightly less than an engine per test cell per day. These engines will burn 8.5×10^6 gallons of jet fuel over a period of one year or about 10^{12} BTU of energy expended. About 80% of this energy is exhausted, because the energy used to operate the turbine is used to operate the compressor, and does not leave the flow. 8×10^{11} BTU/year are, therefore, available for recovery each year, or 4.15×10^9 BTU/workday. If this energy were available on a continuous basis, it would translate to 26.7 megawatts of total power. If 75% recovery were possible and the energy were used to produce electrical power at 20% efficiency, 4 megawatts or 20% of Tinker AFB total electrical demand would be met.* In addition, if 70% of the energy rejected in the energy conversion process were used for low quality heating purposes in place of natural gas, 14% of Tinker AFB total gas demand would be alleviated, since a significant portion of the gas is used in low temperature heat applications.

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* Based upon 1976 energy usage.

The energy source is not, however, continuous. With ten engines per day being tested, an average of .833 per cell, each cell rejects 2.6×10^8 BTU/day. This energy is delivered in a 2.5 hour time frame, however, and during that time, 30 megawatts of power are being rejected. This can be viewed as a 30 megawatt pulse of high quality energy for 2.5 hours and zero megawatts for the remainder of the day. Suppose one were to install one recovery unit per cell, each designed to recover 75% of the exhausted energy. It would then be necessary to design a 23 megawatt recovery system even though the continuous duty rate is 1.67 megawatts (1.74 hours/day) or a workday duty rate of 2.4 megawatts. Because one could expect the cost of the recovery system to be directly proportional to the recovery requirement, individually located units would effectively cost twelve times the cost of a single unit operating continuously. Although this seems to indicate that a central recovery unit is more economical, one must also consider the required building modifications necessary for a central unit. In the case of individual recovery systems, one could still expect to have a central storage and transfer system.

At Tinker AFB, two test cell facilities are currently being operated. One facility has eight test cells, and the other four. The facility with the four cells was recently completed and is specifically designed to test the TF-30 engine, which comprises 28% of the engines tested. The J-75 engine is also tested in the new facility, depending upon the work load. During the period of one year, the number of engines tested would be equally divided between the two facilities; that is, 1/3 of the engines being tested are tested in the new (four cell)

facility and the remainder (2/3) in the old (eight cell) facility. Although the facilities are located across the street from one another, one would expect that two recovery units would be needed because an average of 1.88 engines are being tested continously during the workday. The probability of an engine operating at any one time is .63 in the 4-test cell building and 1.25 in the 8-test cell. This means that 63% of the time an engine will be operating in the 4-test cell building, while in the 8-test cell building at least one engine will be operating virtually all of the time.

Another consideration of the exhaust conditions needs to be noted. Of the 30 megawatts of power being rejected during an engine run, about 20% (6 megawatts) is in the form of mechanical energy. Mechanical energy is directly transferrable as work, and hence, direct energy conversion (DEC) from exhaust kinetic energy to shaft work to electrical energy. DEC usually requires less of a capital investment than high quality heat recovery, and therefore also merits consideration.

Heat recovery from the test cells bears many similarities to heat recovery from a gas turbine; gas turbine recovery systems are operable at many installations and over a wide range of power (some up to 300 megawatts). There are some important differences, however, and these are as follows:

1. As noted previously, the rejected energy is intermittent in nature and occurs at a wide range of power levels.

2. The exhaust possesses a relatively large amount of mechanical energy, most of which is kinetic energy. The exhaust may even be slightly supersonic.
3. The exhaust enthalpy (hence temperature) is slightly higher than in conventional turbine exhausts. Heat exchanger materials may need to be reconsidered.
4. The most significant difference and most likely the most significant constraint is the fact that the augmentor tube must be disconnected from the engine exhaust for adequate engine testing. This not only permits entrainment of ambient air, but proposed recovery equipment could conceivably choke the exhaust flow and cause it to spill out of the augmentor tube intake.

Several levels of the energy recovery are possible, each with special problems and different capital and maintenance costs. The following is a list of potential recovery schemes.

1. Heat recovery of the exhaust gas/entrained air/water spray mixture. In this case, the maximum possible temperature is about 350°F and much of the kinetic energy has been lost due to viscous dissipation.
2. Heat recovery of only the central part of the augmentor tube gases. Such a system will most likely require a separate heat exchanger unit and probably a centrally-located device. In this case, much higher recovery temperatures are possible (800°F), the limiting factor

being that sufficient exhaust kinetic energy must be utilized for entrainment of the ambient air that is required for convective cooling of the tailpipe.

3. Recovery of kinetic energy by a DEC device. In this case, as much of the mechanical energy of the system as possible is extracted with virtually none remaining to drive a heat exchanger.

4. Heat recovery coupled with a DEC device. In this case, either full exhaust gas (1) or central-core (2) recovery can be utilized along with a DEC device, each extracting energy in an optimum manner.

These recovery schemes have been listed in order of increasing difficulty, and most likely, this is directly proportional to the initial capital investment. It is also interesting to note that they are in the order of increasing availability. The amount of useful work that can be produced (i.e., electricity) is directly proportional to availability. The choice of the recovery scheme must be coupled to the potential end use. Obviously, electrical energy with the appropriate switching gear can be directly used by the base. It will, however, require a greater capital outlay to generate and therefore recovery for heat transfer purposes only may prove to be more economically viable providing a suitable use for energy can be found.

14.3 Rejected Energy Recovery from Jet Engine Test Cells: Central Heat Exchanger, Full Exhaust-Gas Utilization

One of the simpler methods of energy recovery from the jet engine test cells involves a centrally located exhauster system which contains a heat exchanger. All the exhaust gases and the entrained air would be ducted directly to a central system as shown in Figure 14.1. The central unit would contain a heat exchanger and the appropriate environmental equipment for noise and air pollution control. The kinetic energy associated with the jet engine exhaust would assist in driving the system. These losses in kinetic energy would exhibit themselves as pressure losses so that the net effect would be an exhaust stack condition of low velocity but ambient pressure.

The major drawback in such a system would be that the temperature would be limited to about 350°F because exhaust gas entrainment dilutes the jet engine exhausts typically to this temperature. None of the thermal energy is lost, but quality of the exhaust energy is significantly reduced. The existing augmentor tubes would be equipped with a sliding collar such that the augmentor tube could be used in its existing mode when the need arises.

Several alternatives exist for the use of the heat exchanger.

a. Hot water for heating application at Building 3001 and other facilities such as the new painting facility.

b. Boiler for production of shaft work.

c. Storage system for multiple use.

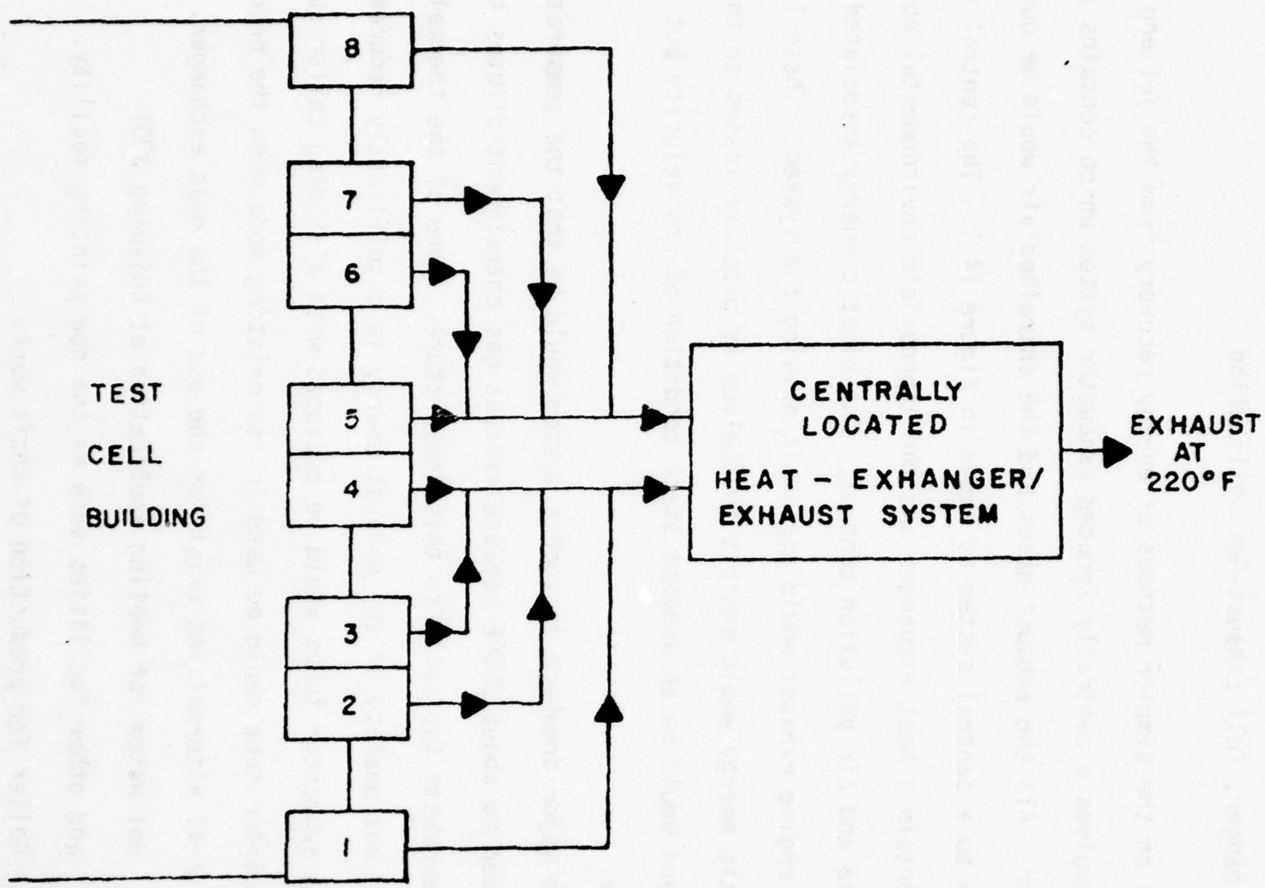

Figure 14.1. Centrally located heat exchanger system for the 8-test cell building at TAFB.

The schematic of a typical system is shown in Figure 14.2. The storage system provides a capitance so that continuous operation of the power generating equipment is possible. The storage requirement is therefore short-term, perhaps on the order of 20 to 30 minutes.

When a jet engine is operating, a heat exchanger directly connected to the organic Rankine-cycle* system is in operation and preheats and boils the working fluid. The storage system fulfills the superheating requirement to raise the working fluid to a temperature of 260°F. When no jet engine is in operation, the storage system provides the entire heating load for the system. The condenser on the Rankine cycle system is shown here to operate at 100°F thus alleviating a cooling tower requirement. The actual Rankine cycle is shown on Figure 14.3 with a 70% efficient pump system and a 91% efficient turbine. Such a unit could be supplied by Sun Power Systems of Miami, Florida. The actual thermal efficiency of the cycle is 12.6% with a net work out of 8.46 BTU/lbm. The required heat input at a maximum temperature of 260°F for the cycle is 67.3 BTU/lbm.

On the other side of the storage system is a heat exchanger system used for providing hot water. It could be used in place of, or in conjunction with, the Rankine-cycle system.

As noted in Section 14.1, an average of 3.15×10^9 BTU/workday was exhausted. Assuming each test cell building is used equally:

8-test cell building	2.10×10^9 BTU/workday
4-test cell building	1.05×10^9 BTU/workday

For energy transfer considerations, relative temperatures are important. The energies above are relative to the reference temperature, for the heat of formation of jet fuel, that temperature being 32°F. Relative to the

*The conditions presented here are for Freon - 12 as the working fluid.

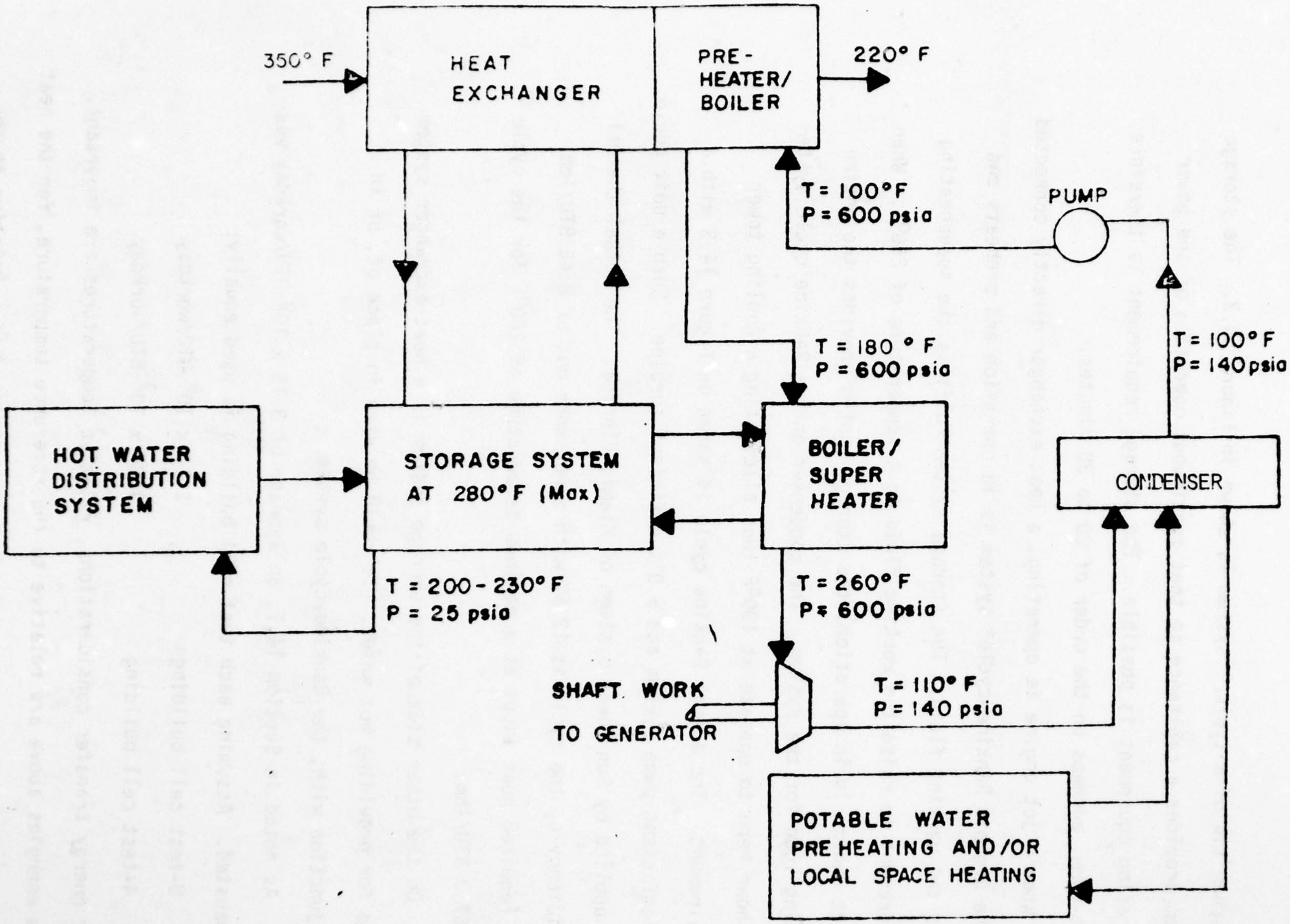

Figure 14.2. Possible recovery system for centrally located heat recovery system

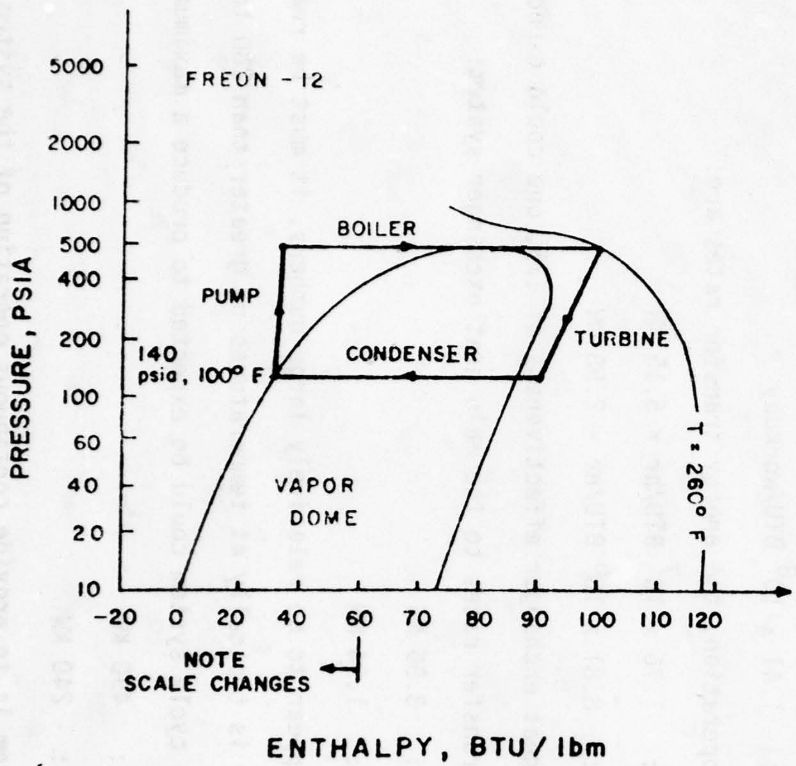

Figure 14.3. Rankine Cycle with Freon-12.

atmospheric temperature in Oklahoma City (59^{o}F), these energies are:

8-test cell: 2.04×10^9 BTU/workday

4-test cell: 1.02×10^9 BTU/workday

In order to minimize condensation and subsequent corrosion in the exhaust stack, the exhaust temperature could be limited to no lower than 220^{o}F. Since the exhaust gas/entrained air mixture is typically 350^{o}F, the amount of energy actually available is represented by a temperature decrease of 130^{o}F or:

8-test cell: 2.81×10^8 BTU/workday

4-test cell: 1.41×10^8 BTU/workday

For 16 hours of operation, the energy transfer rates are:

8-test cell: 1.76×10^7 BTU/hr = 5.15 MW

4-test cell: 8.81×10^6 BTU/hr = 2.58 MW

Assuming a heat exchanger effectiveness of .75, one could expect the following heat transfer rates to the main heat exchanger system:

8-test cell: 3.86 MW

4-test cell: 1.94 MW

Although these appear to be relatively large numbers, it must be remembered that this energy is typically at temperatures no greater than 280 to 300^{o}F.

The Rankine cycle system could be expected to produce a maximum of:

8-test cell: 490 KW

4-test cell: 240 KW

The storage system is to provide continuous operation of the system during the workday. If the storage requirement is for 30 minutes of storage, a pressurized water system could provide it. Storage would be at 280^{o}F, but the Rankine cycle system needs energy at a temperature of no less

than 260°F. The storage requirements are given by the first law of thermodynamics; that is,

$$\text{Energy stored} = E_{STORED} = MC\Delta T \qquad (14\text{-}1)$$

where M is the storage mass, C is the specific heat, and ΔT is the allowable temperature difference. For the conditions herein:

8-test cell:

$$M = \frac{(3.86\ MW)\ (.5\ hr)}{(1\ BTU/lbm\ F^{o})((280\text{-}260)\ F^{o})} = 329{,}000\ lbm$$

Volume = 5300 ft^3

Spherical tank = 22 ft diameter

4-test cell:

M = 220,000 lbm = 27,500 gal.

Volume = 3600 ft^3

Spherical tank = 19 ft diameter

Although none of the requirements are unreasonable, it is disappointing that the level of power available from such a system is so low.

The system that is actually put in practice could have any or all of the components suggested here. The low level of output power in the Rankine cycle suggests that it will not be economical to generate electrical power. The system could be used for the generation of hot water and/or low pressure steam. If steam at 280^{o}F would be generated with this system, it is doubtful that it could be pumped for use at Building 3001 which is about 1/2 mile away.

Therefore, any recovery of energy as heat from this system could only be used in the immediate vicinity. The energy recovered is, however, in excess of building heating and cooling requirements in the immediate vicinity.

14.4 Basic Flow Characteristics of the Jet Engine Augmentor Tube

Several possible recovery schemes were suggested in Section 14.2, and the full exhaust gas utilization for heat recovery only was analyzed in Section 14.3. Before examining other recovery schemes, the basic flow characteristics within the augmentor tube must be examined. The augmentor is quite similar in operation to an ejector and may be analyzed using turbulent jet ejector theory.

The similarities of ejector operation and augmentor tube operation are shown in Figure 14.4. This figure shows the character of the velocity field at the lateral cross-sections of the mixing chamber. The two streams are mixed in such a manner that the nonuniformity of velocity, temperature, and composition at the lateral cross-section gradually decreases, and at a certain cross-section of the chamber (i.e., at the diffuser inlet) there is formed a nearly uniform flow of the mixture of the ejecting (jet exhaust) and the ejected (entrained) air gases. This happens at the expense of a considerable increase in entropy which exhibits itself as a loss in flow velocity through viscous dissipation and a decrease in exhaust temperature through diffusion.

Before addressing the gradients associated with turbulent jets, much may be learned from the examination of final conditions at the end of the ejector where the flow field is nearly uniform. One may expect such conditions to exist at the entrance to the diffuser. Since the system is confined, conservation equations describe the result. It will be assumed that the ejected gas (2) and the ejecting gas (3) (c.f., Figure 14.4)

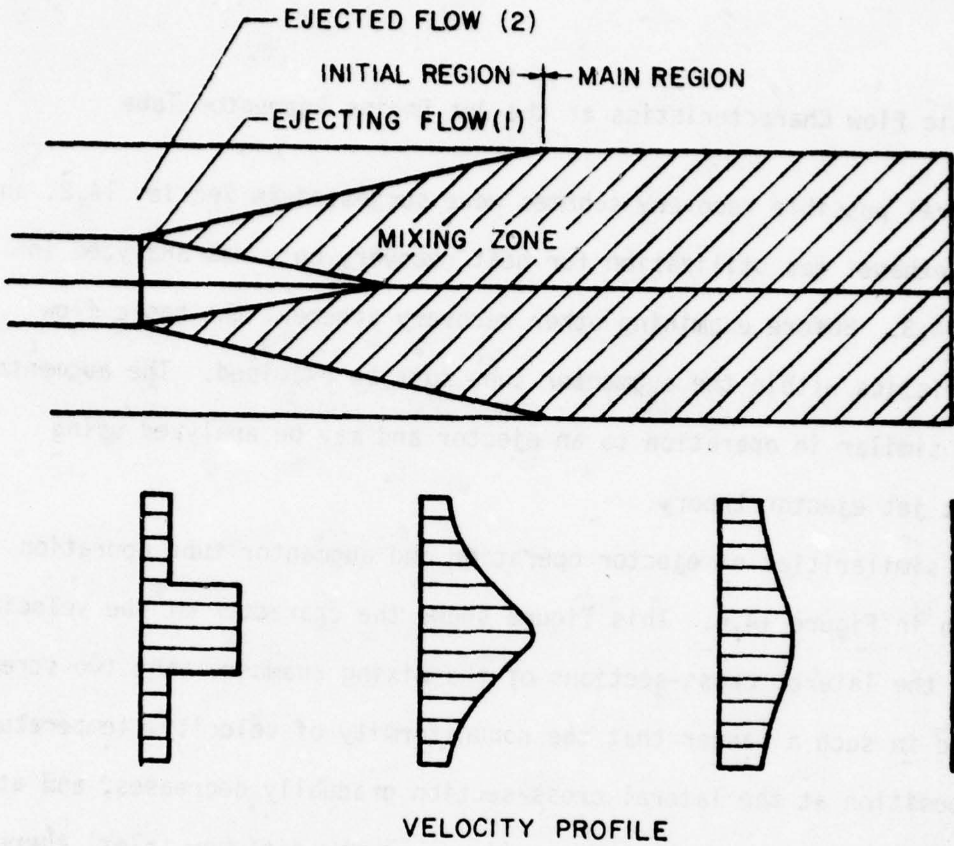

Figure 14.4. **Flow pattern in the mixing chamber of the ejector.**

have similar properties, and are perfect gases. The basic conservation equations are:

Mass: $\dot{M}_1 + \dot{M}_2 = \dot{M}_3$ (14-2)

Momentum: $P_1A_1 + \dot{M}_1U_1 + P_2A_2 + \dot{M}_2U_2 = P_3A_3 + \dot{M}_3U_3$ (14-3)

Energy: $\dot{M}_1(C_pT_1 + \frac{U_1^2}{2}) + \dot{M}_2(C_pT_2 + \frac{U_2^2}{2}) = \dot{M}_3(C_pT_3 + \frac{U_3^2}{2})$ (14-4)

State: $P = \rho RT$ and $\dot{M} = \rho UA$ (14-5)

where $\dot{M}$ is the mass flow rate, P is the pressure, A is the cross-section, U is the axial velocity, C_p is the specific heat at constant pressure, T the temperature, ρ the density, and R the specific gas constant. In each equation the subscripts refer to the various flow states: 1 refers to the ejecting flow (jet engine exhaust), 2 refers to ejected flow (entrained air), 3 refers to flow parameters at the final cross-section of the augmentor tube where complete equalization of all the parameters over the cross-section is assumed.

Consider a typical jet engine test in the eight test cell building. In the eight test cell building, the most common engine being tested is the J-57. The basic parameters for this engine at military power conditions are:

Mass flow rate = $\dot{M}_1$ = 185 lbm/sec

Exhaust gas stagnation temperature = 1100°F

Exhaust gas temperature = T_1 = 880°F

Exhaust gas velocity = U_1 = 1800 ft/sec

Exhaust area = A_1 = 3.65 ft^2

Exhaust pressure = $P_1 = P_2$ = 14.3 psia

Exhaust density = $\rho_1 = 8.76 \times 10^{-4}$ slugs/ft^3

The augmentor tube conditions are:

Inside area $= A_3 = 28.3\ ft^2$

Ejected area $= A_3 - A_1 = A_2 = 24.6\ ft^2$

Ejected temperature $= T_2 = 60^oF$

Ejected density $= 2.31 \times 10^{-3}\ slug/ft^3$

Table 14.3 indicates the corresponding conditions at 2 and 3 for various augmentation ratios (k). These results have been obtained through the solution of the conservation equations. For the J-57 the temperatures measured at the diffuser are on the order of 300 to 350°F, corresponding to an augmentation ratio of 3.0 to 2.5, respectively. The entrained mass flow rate is therefore, on the order of 3 times the mass flow rate of the jet exhaust. Note also that there is a slight increase in pressure. These computations do not reflect frictional losses along the augmentor tube walls. Although these losses will be small compared to total flow energy, they contribute to a loss in pressure. The assumption, therefore, that the pressure is nearly constant along the augmentor is a good one.

Experimental investigations have established an interesting analogy between the velocity fields at the lateral cross-sections of a mixing chamber and at the cross-sections of a free jet. It was found that the process of equalization of the flow parameters in a cylindrical mixing chamber occurs in such a manner that the velocity field at each of its cross-sections (cf. Figure 14.5) appears as if it were the central part, bounded by the cylindrical walls of the chamber, of the universal function, which expresses the dimensionless velocity field at the corresponding

k	$\dot{M}_1$ lbm/sec	$\dot{M}_2$	$\dot{M}_3$	U_2(ft/sec)
0	185	0	185	0
0.5	185	93	278	53
1.0	185	185	370	107
1.5	185	278	463	160
2.0	185	370	555	213
2.5	185	463	648	266
3.0	185	555	740	320
3.5	185	648	833	373
4.0	185	740	925	426

k	ρ_3 slug/ft^3	P_3(psia)	T_3(°F)	U_3(ft/sec)	$\dot{E}_K$(MW)
0	8.96×10^{-4}	16.6	1094	239	.24
0.5	1.14×10^{-3}	16.4	745	282	.49
1.0	1.32×10^{-3}	16.1	572	326	.87
1.5	1.45×10^{-3}	15.9	462	370	1.4
2.0	1.55×10^{-3}	15.7	393	415	2.1
2.5	1.63×10^{-3}	15.5	340	460	3.0
3.0	1.69×10^{-3}	15.4	304	506	4.2
3.5	1.74×10^{-3}	15.2	272	553	5.7
4.0	1.78×10^{-3}	15.0	247	601	7.4

Table 14.3. Flow characteristics in the augmentor tube.

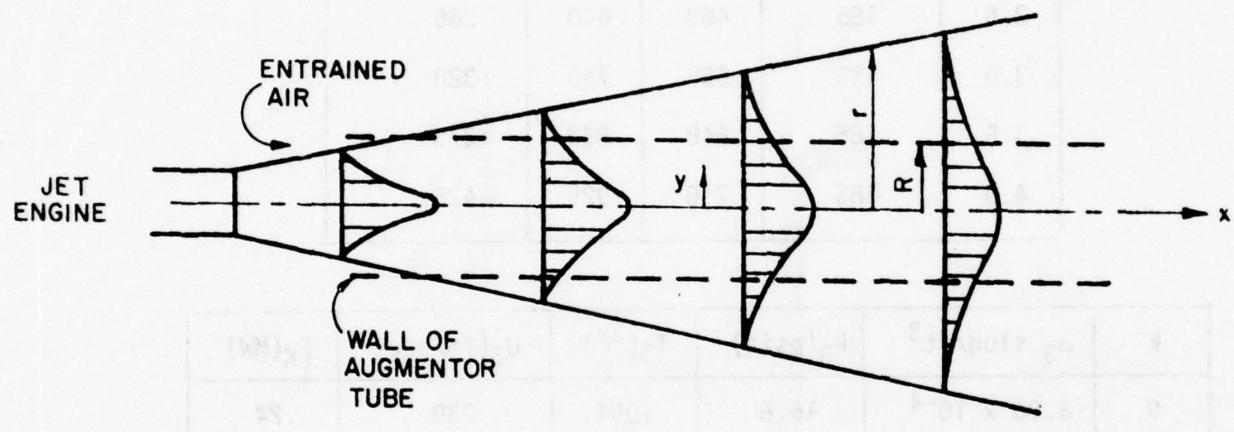

Figure 14.5. The analogy between the fluid flow of a free jet and a confined jet.

cross-section of a free jet. This analogy between the velocity fields in the ejector and in a free jet is explained, obviously, by the universality of the effective laws of turbulent mixing.

As in the study of the free jet, the existence of this universality of the field at different cross-sections of the flow makes it possible to set up integral equations which express the fundamental laws of conservation of mass momentum and energy, and to define the flow parameters at any arbitrary point of the mixing chamber in terms of the initial parameters of the mixing streams. It is convenient to define a nondimensional parameter, $\xi = y/r$, where y and r are as shown in Figure 14.5, y being the radius to some point and r the radius of the free jet at the same cross-section. For a confined jet the parameter ξ becomes ξ_k when y = r where R is the radius of the augmentor tube.

The basic flow considerations for central core recovery are established from the results of typical ejector flow field analysis. It proceeds from the analogy between the flow in a turbulent free jet and the flow in a mixing chamber of the ejector. The velocity distribution of a subsonic turbulent free jet at any axial position x has been determined through experiments [12] to be very nearly

$$\frac{u}{u_m} = (1 - \xi^{1.5})^2 \tag{14-6}$$

where u is the velocity in the x-direction and u_m is the velocity on the axis at that x location in the flow. The temperature distribution is related to the velocity distribution as follows:

$$\frac{T - T_2}{T_m - T_2} = \sqrt{\frac{u}{u_m}} = 1 - \xi^{1.5} \tag{14-7}$$

The evaluation of the centerline temperature, T_m as a function of x can be accomplished downstream of the initial mixing region if the kinetic energy is assumed to be negligible compared to the "absolute" thermal energy of the flow. To assess the validity of this assumption, consider the thermal and kinetic energy at the jet engine exhaust and at station 3. At the exhaust plane,

$$\frac{KE}{TE} = \frac{\frac{U_2^2}{2}}{C_p T_2} = \frac{\left(1800 \frac{ft}{sec}\right)^2}{(2)\left(6000 \frac{ft^2}{sec^2 °R}\right)(1340°R)} = 20\%$$

at station 3 for an augmentation ratio of 3.0,

$$\frac{KE}{TE} = \frac{\frac{U_3^2}{2}}{C_p T_3} = \frac{\left(506 \frac{ft}{sec}\right)^2}{(2)\left(6000 \frac{ft^2}{sec^2 °R}\right)(764°R)} = 2.8\%$$

Although the kinetic energy is a measurable fraction of the energy at the exhaust plane, it degrades to small value at the fully-mixed equilibrium location. It can be assumed, therefore, that errors in temperature of no more than 10% will result by neglecting kinetic energy changes from the incipient fully mixed case ($\xi_k = 1$) to the fully-mixed equilibrium location (3). This assumption reduces the integral energy equation (14-3) to:

$$\dot{M}(C_p T) = \dot{M}_3(C_p T_3) \qquad (14\text{-}8)$$

Downstream of incipient full mixing, the mass flow rate is constant and, assuming constant C_p, the average temperature is constant. That

is:

$$\overline{T} \approx T_3 \tag{14-9}$$

Using this assumption and equation (14-7) for the temperature distribution, a relationship for T_m as a function of ξ_k and hence x may be found. The average temperature is:

$$\overline{T} = \frac{2\pi\int_0^R T(y)\, y dy}{2\pi\int_0^R y dy} \tag{14-10}$$

for the distribution in equation (4-8):

$$\overline{T} = \frac{2}{R^2}\int_0^R [(T_m - T_2)\,(1 - \xi^{1.5}) + T_2]\, y dy \tag{14-11}$$

for $\xi = \frac{y}{r}$ and $\xi_k = \frac{R}{r}$,

$$\overline{T} = \frac{2}{\xi_k^{\,2}} \int_0^{\xi_k} [(T_m - T_2)\,(1 - \xi^{1.5}) + T_2]\, \xi d\xi \tag{14-12}$$

Integrating:

$$\overline{T} = \frac{2}{\xi_k^{\,2}} [(T_m - T_2)\,[\frac{\xi_k^{\,2}}{2} - \frac{\xi_k^{\,3.5}}{3.5}] + T_2 \frac{\xi_k^{\,2}}{2}] \tag{14-13}$$

Simplifying:

$$\overline{T} = (T_m - T_2)\,(1 - .571\, \xi_k^{\,1.5}) + T_2 \tag{14-14}$$

Using the result of equation (14-11) and solving for $T_m - T_2$:

$$T_m - T_2 = \frac{T_3 - T_2}{1 - .571\,\xi_k^{1.5}} \qquad (14\text{-}15)$$

Consider a central core recovery scheme as shown in Figure 14.6. A recovery tube of diameter equal to the engine is placed in the augmentor at the location where the jet is just fully mixed ($\xi_k = 1$). The maximum temperature from equation (14-15) for an augmentation ratio of 3 is:

$$T_m = \frac{304°F - 60°F}{1 - .571} + 60°F \qquad (14\text{-}16)$$

$$T_m = 629°F = 1089°R$$

The temperature distribution is then

$$T = 569F° \,(1 - \xi^{1.5}) + 60°F \qquad (14\text{-}17)$$

Since $\xi_k = 1$, $r = R$, the average temperature in the recovery tube (diameter 2.2 ft, $\xi_r = 0.36$),

$$\bar{T} = (569F°)\,(1 - .571\,\xi_r^{1.5}) + 60°F \qquad (14\text{-}18)$$

$$\bar{T} = 559°F$$

The recovery tube would therefore be capable of capturing energy at a much higher temperature than in full recovery. Some energy is lost in the remaining augmentor tube region, however. The mass flow rate in the recovery tube may be approximated to lowest order to be equal to the mass

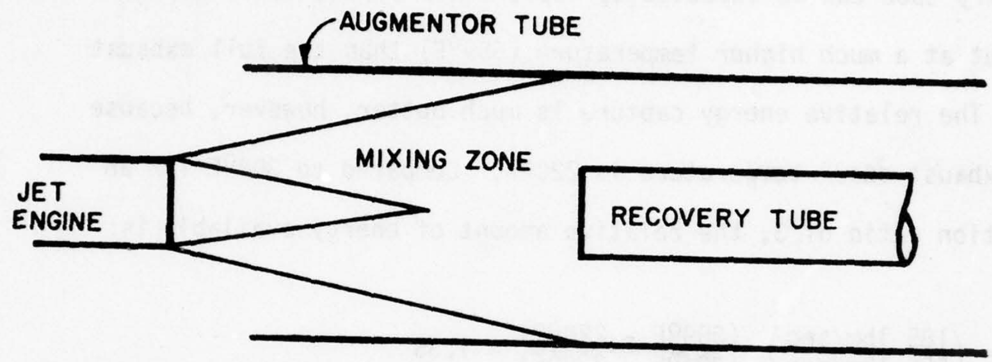

Figure 14.6. Possible recovery scheme for central-core high temperature energy recovery.

flow rate out of the jet engine. In this case, the recovered energy fraction is:

$$\eta_1 = \frac{\dot{M}_r(C_p\bar{T})}{\dot{M}_3(C_pT_3)} = \left(\frac{185 \text{ lbm/sec}}{555 \text{ lbm/sec}} \times \frac{1019^oR}{764^oR}\right) = .44 \tag{14-19}$$

The recovery tube can be expected to capture 44% of the total thermal energy, but at a much higher temperature (559°F) than the full exhaust capture. The relative energy capture is much better, however, because minimum exhaust stack temperature is 220°F. Compared to 304°F for an augmentation ratio of 3, the relative amount of energy available is:

$$\eta_2 = \left(\frac{185 \text{ lbm/sec}}{555 \text{ lbm/sec}}\right) \frac{(559^oF - 220^oF)}{(304^oF - 220^oF)} = 1.35 \tag{14-20}$$

The result of equation (14-20) indicates that although some energy must be given up to operate the augmentor tube, the availability of the recovered energy remains high and still more energy can potentially be recovered.

With possible recovery temperatures of over 500°F, the complexion of the recovery system changes. It should be possible to generate high-temperature hot-water (>400°F) that could be distributed to various locations. In addition, a more efficient Rankine-cycle could operate for power generation if desired. An organic Rankine-cycle with freon-12 as the working fluid and isentropic efficiencies similar to those in Section 4.3, but operating at 450°F instead of 280°F would be capable of a thermal efficiency of 21% with a 21 BTU/lbm energy output across the compressor. Such a system with proper storage could deliver 0.7 Mw of shaft power.

Central core recovery can provide a significant advantage for production of steam. From the shaft power generation aspect, however, it appears that the electrical generation potential still seems marginal. The possibility of generating high temperature hot water for a closed high temperature hot water distribution system is also attractive.

14.5. Direct Energy Conversion from a Jet Engine Exhaust by Mechanical Means

As noted in the previous section, considerable kinetic energy flux remains even after complete mixing occurs within the augmentor tube. This number is a relatively small percentage of the total energy available in the flow but because of its magnitude, (greater than 1 Mw), it merits further consideration. The problems associated with direct energy conversion by mechanical means are addressed in this section.

Before considering the arrangement of optimum energy capture for DEC in an augmentor tube-type configuration, consider the basic phenomena associated with conventional windmills. A conventional unaugmented windmill with the relevant flow variables is shown in Figure 14.7. Figure 14.7a indicates the stream tube for optimum energy capture. Figures 14.7b and 14.7c indicate the velocity and pressure profiles for optimum recovery of energy from the wind. Because the recovery involves air at free stream pressure, not all of the kinetic energy can be extracted from the momentum transfer process. This results in the conventional .593 value for windmill efficiency. The decrease in the velocity is a continuous process while the pressure jumps discontinuously across the windmill. The process involves the transfer of dynamic pressure to static pressure, energy extraction, and decrease of dynamic pressure to return static pressure to ambient conditions. The stream tube continually expands in order that the mass flow may remain constant as the velocity decreases.

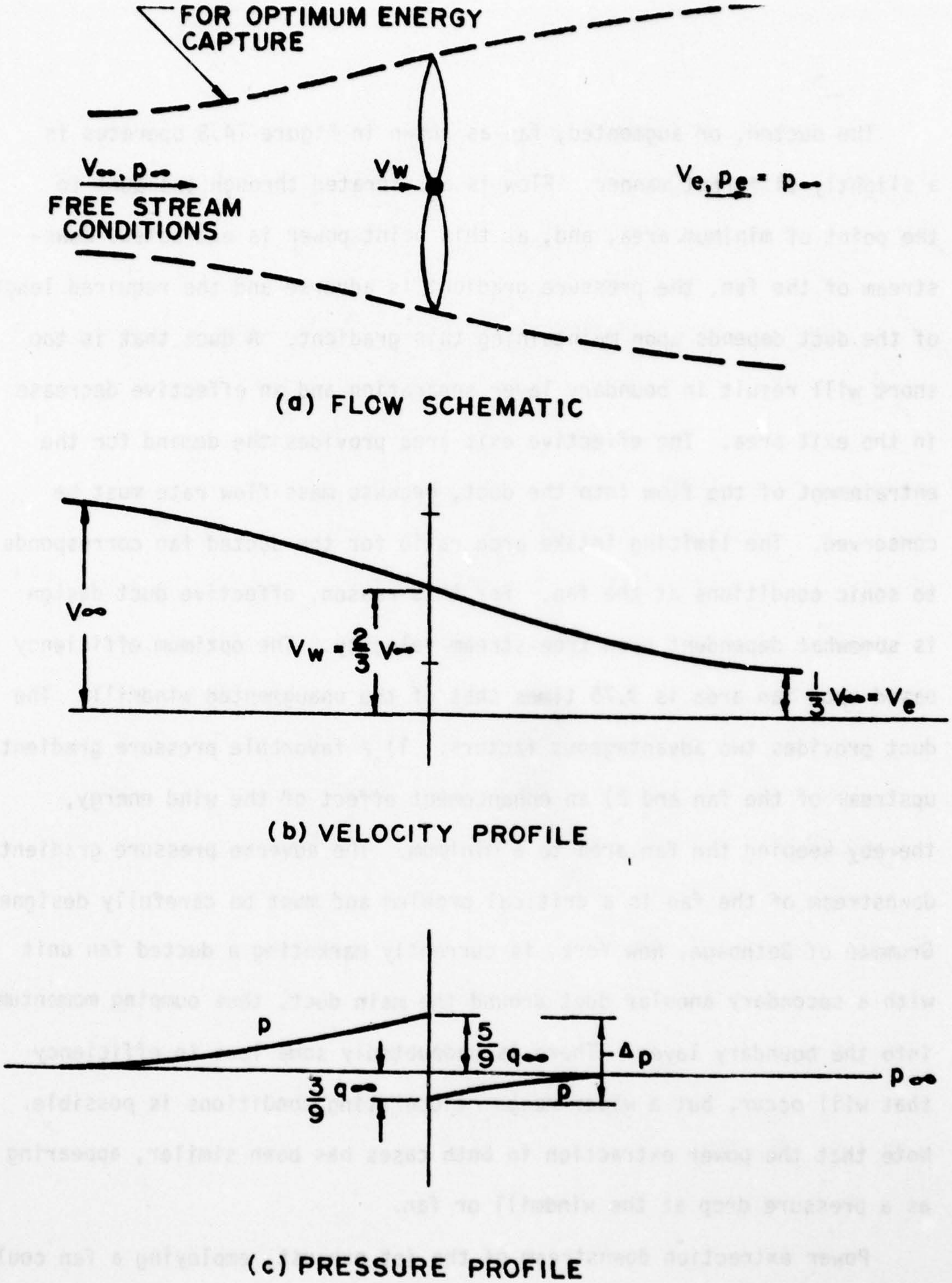

Figure 14.7. Basic processes associated with optimum extraction of wind energy by an unaugmented windmill.

The ducted, or augmented, fan as shown in Figure 14.8 operates in a slightly different manner. Flow is accelerated through the duct to the point of minimum area, and, at this point power is extracted. Downstream of the fan, the pressure gradient is adverse and the required length of the duct depends upon maintaining this gradient. A duct that is too short will result in boundary layer separation and an effective decrease in the exit area. The effective exit area provides the demand for the entrainment of the flow into the duct, because mass flow rate must be conserved. The limiting intake area ratio for the ducted fan corresponds to sonic conditions at the fan. For this reason, effective duct design is somewhat dependent upon free-stream velocity. The optimum efficiency based upon fan area is 3.75 times that of the unaugmented windmill. The duct provides two advantageous factors: 1) a favorable pressure gradient upstream of the fan and 2) an enhancement effect of the wind energy, thereby keeping the fan area to a minimum. The adverse pressure gradient downstream of the fan is a critical problem and must be carefully designed. Grumman of Bethpage, New York, is currently marketing a ducted fan unit with a secondary annular duct around the main duct, thus pumping momentum into the boundary layer. There is undoubtedly some loss in efficiency that will occur, but a wider range of operating conditions is possible. Note that the power extraction in both cases has been similar, appearing as a pressure drop at the windmill or fan.

Power extraction downstream of the jet exhaust, employing a fan could occur, similar to the design shown in Figure 14.8. The jet exhaust presents some unique flow phenomena that must be considered in the design.

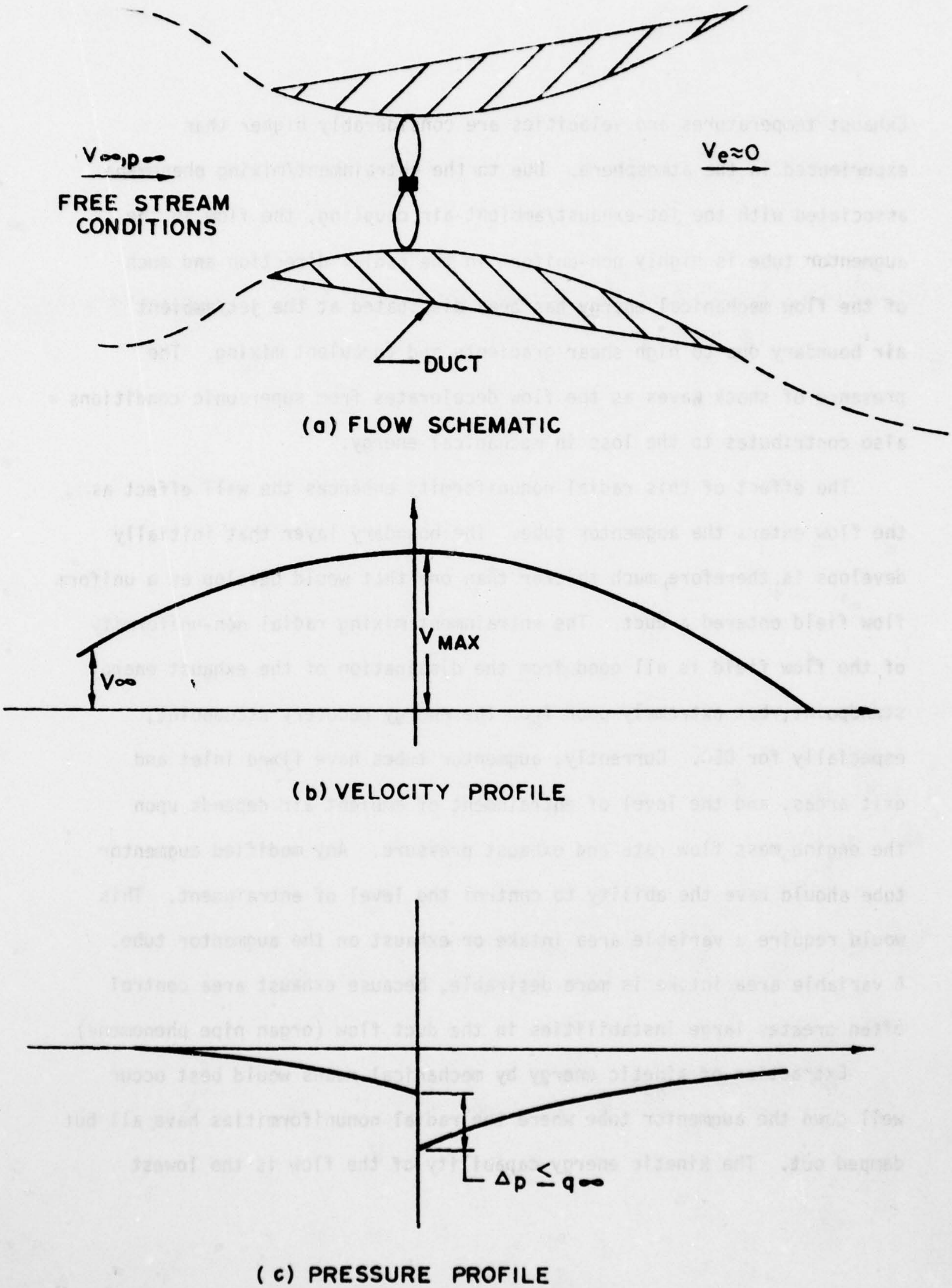

Figure 14.8 Basic processes associated with a ducted windmill operating in a low velocity wind.

Exhaust temperatures and velocities are considerably higher than experienced in the atmosphere. Due to the entrainment/mixing phenomena associated with the jet-exhaust/ambient-air coupling, the flow in the augmentor tube is highly non-uniform in the radial direction and much of the flow mechanical energy has been dissipated at the jet/ambient air boundary due to high shear gradients and turbulent mixing. The presence of shock waves as the flow decelerates from supersonic conditions also contributes to the loss in mechanical energy.

The effect of this radial nonuniformity enhances the wall effect as the flow enters the augmentor tube. The boundary layer that initially develops is, therefore, much thicker than one that would develop as a uniform flow field entered a duct. The entrainment mixing radial non-uniformity of the flow field is all good from the dissipation of the exhaust energy standpoint, but extremely poor from the energy recovery standpoint, especially for DEC. Currently, augmentor tubes have fixed inlet and exit areas, and the level of entrainment of ambient air depends upon the engine mass flow rate and exhaust pressure. Any modified augmentor tube should have the ability to control the level of entrainment. This would require a variable area intake or exhaust on the augmentor tube. A variable area intake is more desirable, because exhaust area control often creates large instabilities in the duct flow (organ pipe phenomena).

Extraction of kinetic energy by mechanical means would best occur well down the augmentor tube where the radial nonuniformities have all but damped out. The kinetic energy capability of the flow is the lowest

here, but sufficient power remains. As noted in Table 14.3, 3 to 4 megawatts exist and are available in a J-57 test. Even at 25%, this is still .75 to 1 megawatts of power. Power extraction would be limited by the pressure requirements downstream. If heat recovery equipment is used, some kinetic energy must remain in the flow for appropriate pressure losses across the heat recovery equipment.

If the entrainment level is decreased, as is the case for large engines, more kinetic energy remains in the flow requiring a different duct design. The reason for this is that the flow is near sonic velocity. Care must be taken to keep the flow velocity below the sonic point, since local supersonic conditions could then occur on the fan blades, creating significant problems not only in power extraction, but structurally as well. Therefore, as entrainment level is decreased, one would expect the converging section to decrease. In the limit of zero entrainment, the requirement is more complex. From the mechanical energy standpoint, a low entrainment flow field would most likely deliver about the same energy as a moderate entrainment flow field, the losses being caused by different mechanisms.

The fan employed for power extraction must also be constructed of materials capable of withstanding the high temperatures that exist in a jet exhaust. This problem is not too severe, because the turbine in the jet engine is made to withstand even more severe conditions. The design of a turbine to withstand afterburner temperatures would approach the limits of the state-of-the-art in materials. Therefore, if DEC devices are employed in test cells, at least one test cell will have

to remain free for afterburner testing. DEC recovery with fans will most likely have to be excluded from Air Force bases with only one test cell or those which primarily test afterburning engines.

The recovery of mechanical energy from the jet engine exhaust presents an arduous task for DEC conversion. Potential losses due to skin friction and flow turning make a centrally located DEC unit impractical. If no DEC is used, this energy is not lost because this high momentum flow permits more freedom in the design of the heat recovery unit. In addition, because a jet exhaust is a compressible flow field, the flow decelerates for entry into a heat recovery unit, the energy appears in the enthalpy, and, temperature can subsequently be recovered. It is not, however, transferable as work and is limited by the Carnot efficiency for the production of work.

The potential recovery of energy in direct energy conversion is about a factor of two greater than the case where work is produced from the heat recovery system discussed in the previous section. The basic cost of a rankine cycle unit and a ducted windmill could be expected to be the same, with about $1 per watt of shaft power generated, not including the appropriate switching equipment. DEC units would have to be individually located, however, and the relatively low utilization time will probably result in a long break-even period.

14.6 The Prospect for Electrofluidynamic Power Generation in the Jet Engine Test Cell

An Electrofluidynamic (EFD) power generator is a direct energy conversion device which converts fluid dynamic energy into electrical energy. In its simplest form, the EFD* generator consists of ions, usually a corona discharge, and a downstream collector of these ions. This is shown schematically in Figure 14.9. The device is analagous to a Van de Graaff generator except that the belt is replaced by a gas stream. Once the gas is seeded with ions, they are transported downstream against the action of an opposing electric field and the gas must go to work as it forces the ion current downstream and through the external load. From a microscopic view, the opposing electric field produced by the current through the external load exerts an electrostatic force on the ions in the gas. These ions, in turn, collide, transmitting momentum and energy to the gas flow. In other words, the electrostatic force on the ions is transmitted undiminished to the gas and some of the gas kinetic energy is directly converted into electric energy.

EFD power generation, like magnetogasdynamic (MGD) power generation, is a dynamic, direct energy conversion process in which the energy of

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*The term electrofluid dynamic power generation is usually used in cases when the flow is incompressible and electrogasdynamic (EGD) power generation is used for compressible flow cases.

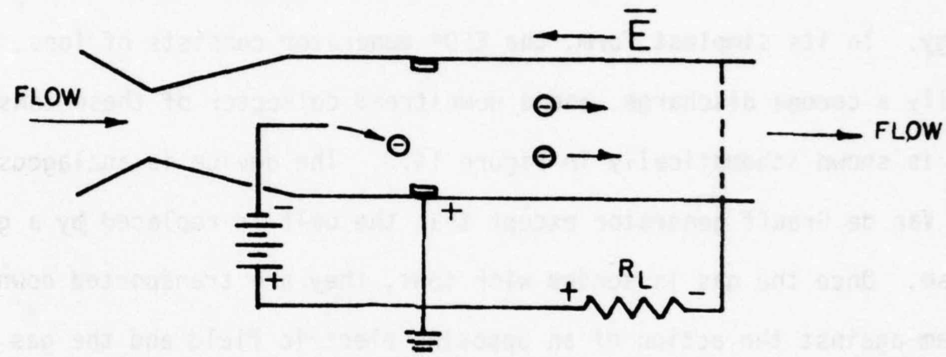

Figure 14.9. Simple EFD generator

flowing gases is converted into electricity without moving mechanical parts. In principle, this offers the possibility of generating electricity at higher efficiency, because higher temperature working fluids can be used. MGD generators are very low internal impedance generators, while EFD generators are extremely high internal impedence generators. Thus, EFD generators have an output which is at a very high voltage (100 KV to 1000 KV) and a very low current. Conversion to low voltage power is, therefore, necessary, thus increasing the cost of the system. EFD power generation has the advantage over MGD power generation-it requires no magnet for implementation. MGD power generation is apparently favored for direct energy conversion schemes because the losses associated with EFD usually are more severe. Only where weight is of primary importance does EFD show an advantage. The losses associated with MGD power generation severely limit the practicability of small MGD generators; the most promising use therefore appears to be in conjunction with large central power stations, and plants in 100 Mw range are being studied. Since it is a low impedance operating device, the ability to maintain electrical conductivity is severely limited by a minimum operating temperature range. Thus, MGD is limited to high enthalpy flow with temperatures in excess of 2000^{o}R being the apparent minimum practical value. Due to these facts, MGD has no apparent practicality at atmospheric pressures and the temperatures and velocities associated with the typical exhaust conditions at a jet engine test cell. Jet engine exhaust conditions are degraded further due to the required entrainment and mixing of ambient air that aids in tail pipe cooling. EFD, however, can operate at these conditions; and, as will be noted, other secondary advantages and benefits may be derived from its use.

EFD generators are normally considered to operate in homogeneous media and at high pressures, such as 100 atmospheres, to inhibit arcing problems. Special problems will, therefore, exist that are germane only to this situation. Historically, EFD generators have exhibited several problems: (1) ion slip losses; (2) friction losses; (3) ion dispersion losses; and (4) breakdown limitations. Gas seeding is commonly used to minimize the severest of these problems — ion slip losses. The gas is usually seeded with aerosol droplets which have a much higher drag coefficient than a single molecule or atom, thus minimizing slip losses. In the case of jet exhaust in the test cell, particulates exist and water can easily be implemented as a charge carrier if necessary. Small charge carriers have high mobility, but large aerosols or colloidal particles have low mobility. To be efficient, however, slip must be minimized and this requires aerosols.

A schematic diagram of a simple EFD generator in a redesigned augmentor tube is shown in Figure 14.10. The generator system consists of the following:

a) a method for producing charged colloids

b) an inlet electrode which also serves as an attractor electrode

c) a collector electrode guide

d) a high voltage power supply

e) a control system

f) a transfer system to a central load control unit

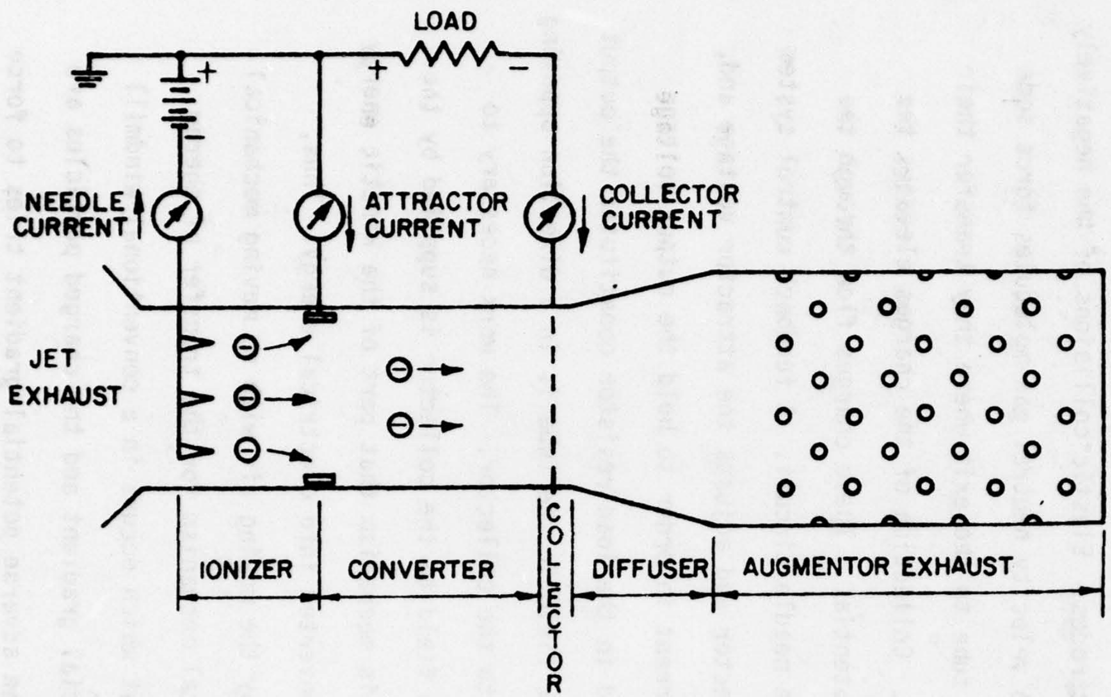

Figure 14.10 Schematic of EFD generator contained in the redesigned augmentor tube.

The high voltage power supply places the attractor electrode at a high voltage relative to the colloid producing system. Charged colloid particles of one sign (usually negative) produced by the charging system, drift toward the attractor electrodes. Elastic collisions of the negatively charged particles with the high velocity neutral gas molecules force some of these ions to flow down the tube to the exit where they transfer their charges to the metal collector. Collection of the charges elevates the collector to a high negative potential. These charges flow through the load resistor, returning to the needle circuit. A feedback control system senses the voltage on the collector and adjusts the attractor voltage and, thereby, controls the output current in order to hold the output voltage constant. The power dissipated in the load resistor constitutes the output of the generator. The electric field in the tube is in a direction opposing the flow of charged particles to the collector. The work necessary to transport the ions against the field to the collector is supplied by the neutral gas. It is through this mechanism that part of the kinetic energy of the high velocity gas is converted into electrical energy. Thus, electrical power is produced by the moving air with no moving mechanical parts required. The fundamental mechanism for the transfer of energy from the gas is similar to that which occurs in a conventional windmill except that the adverse potential gradient and the charged particles are analogous to the windmill. The adverse potential gradient tries to force the particles in the opposite direction of the air flow. The flow does work against this potential and drives the particles downstream. The action due to pressure in the form of drag on the charged particles appears

as current through the load generative power. The pressure drop created by this process limits the magnitude of the applied voltage. At atmospheric pressures, breakdown considerations limit the pressure drop to about .0058 psi. The slightly lower pressure now requires a diffuser on the augmentor tube so that the flow can be returned to atmospheric pressure. If this is not accomplished, then the augmentor will adjust itself upstream to reduce the inlet mass flow. The net result is an unstable system which may result in some rather large oscillatory pulses of pressure and mass flow in the augmentor. If the jet exhaust and entrainment process is hampered, severe problems, involving both the engine structure and the safety of the operating personnel will occur.

The theory of EFD devices is well developed and scaling laws are understood. Experimental evidence compares favorably with available theory and the feasibility of the EFD generator has been well established. The breakdown limitation presents the severest constraint on the system. The ideal windmill extracts 8/9 of the dynamic pressure from the wind, and 0.0058 psia corresponds to 19.2 mph. Below 19.2 mph, the performance of an EFD generator is not limited by electrical breakdown and should operate similar to a conventional windmill. In the augmentor tube, however, one can expect velocities of 300 to 400 mph at five to ten diameters down the augmentor tube from the entrance.

In this region, the pressure drop would need to be fixed at 0.0058 psi and the power output would, therefore, increase linearly with the velocity through the conversion section. The power output of a conventional windmill varies at the cube of the velocity until the design condition is reached, above which it becomes a constant with respect to velocity. The

resulting performance comparison is shown in Figure 14.11. Included in the graphs is the effect of slip for a particle mobility of 10% (ft/sec) (V/m) which can be achieved in practice. The breakdown limit of 0.0058 psi assumes a one-dimensional uniform flow field. The radial nonuniformities caused by the jet exhaust will result in higher attainable pressure drops, and steepen the performance curves. Figure 14.11 is particularly disturbing since the practical EFD generator at 300 mph can extract only 26% of the ideal wind generator performance.

This efficiency must be tempered with the cost of an EFD generator (about \$10 - \$25/ft^2), if one excludes its switching circuitry to tie into the power lines. For a 10 ft diameter augmentor tube, this translates to \$785 to \$2000; the practical EFD generator could deliver about 214 watts/ft^2 or 16.8 kw, or \$40 to \$120/KW. Therefore, although the efficiency is very low, the cost per kilowatt is quite attractive. In addition, the EFD generator is capable of operating as a precipitator. One could, therefore, view the system as a precipitator which has a side performance characteristic of wind energy conversion.

There are two major causes for concern with the implementation of EFD; one being the requirement for energy storage or rapid switching, and the other being the requirement for a small pressure change to avoid breakdown. Although power switching equipment can meet the rapid switching requirements for a jet engine test, it was never intended to operate in that manner.

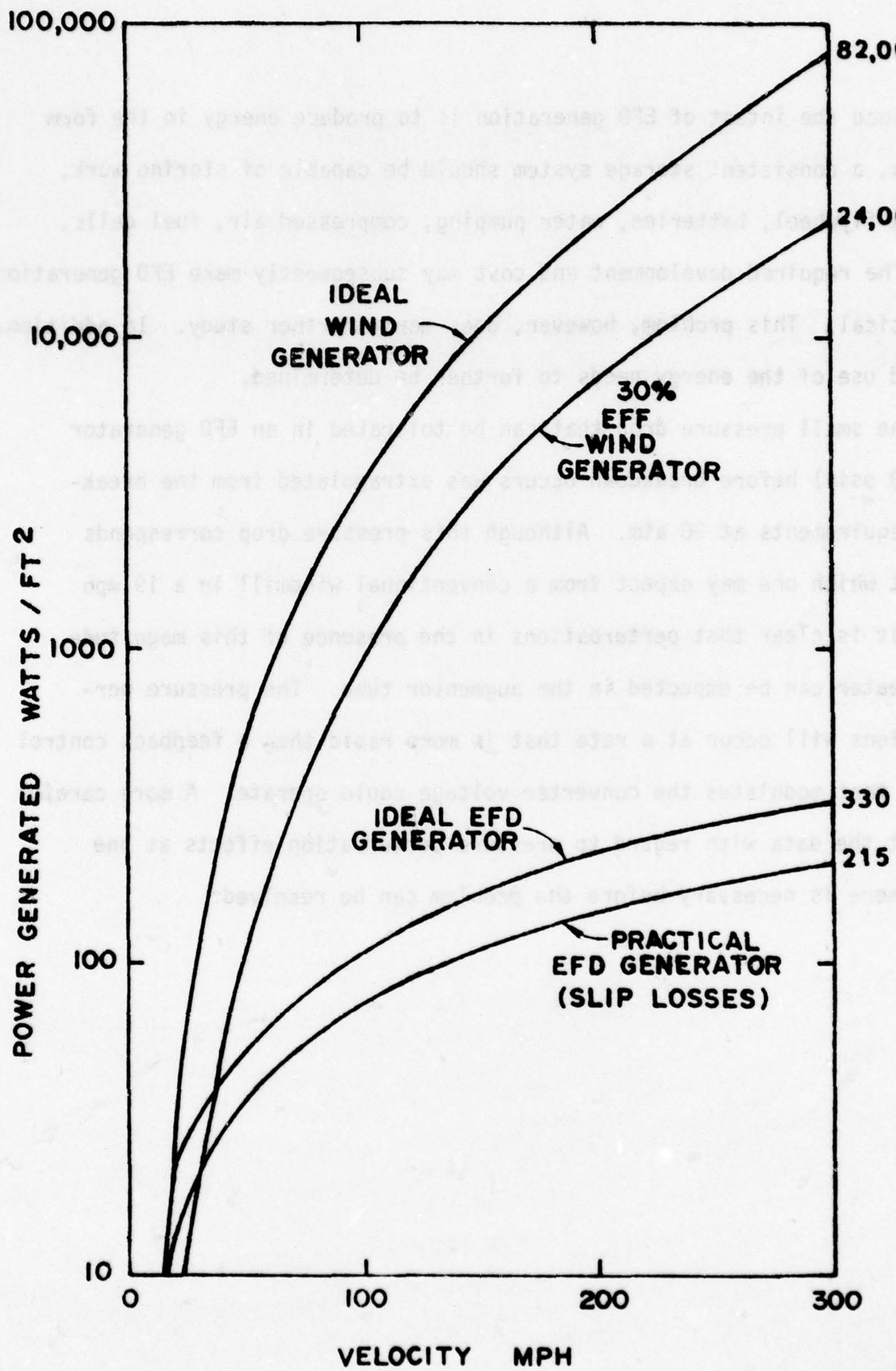

Figure 14.11. EFD - Wind generator performance comparisons.

Since the intent of EFD generation is to produce energy in the form of work, a consistent storage system should be capable of storing work, i.e., a flywheel, batteries, water pumping, compressed air, fuel cells, etc. The required development and cost may subsequently make EFD generation impractical. This problem, however, does merit further study. In addition, the end use of the energy needs to further be determined.

The small pressure drop that can be tolerated in an EFD generator (0.0058 psia) before breakdown occurs was extrapolated from the breakdown requirements at 30 atm. Although this pressure drop corresponds to that which one may expect from a conventional windmill in a 19 mph wind, it is clear that perturbations in the presence of this magnitude and greater can be expected in the augmentor tube. The pressure perturbations will occur at a rate that is more rapid than a feedback control system that modulates the converter voltage could operate. A more careful look at the data with regard to pressure perturbation effects at one atmosphere is necessary before the problem can be resolved.

14.7 Basic Electrical Equipment Necessary for the Waste Heat Utilization from the Jet Engine Test Cells

The electrical generating system will use the Rankine Cycle engine as the prime mover, with an appropriate gearbox to match the generator's required shaft rotation speed of 1800 RPM.

Typical of the small-scale power generators which could be specified for the jet engine test-cell facility energy recovery is a brushless generator manufactured by General Electric Corporation, Schenectady, New York, available in several models, whose ratings cover the range from 150 to 3500 KW.

Any of these models can be made to match the Base's primary distribution voltage by means of appropriate step-up transformers, since the voltage output of this model series is 3-phase at 60 HZ. This model series is fully ruggedized and is an element of GE's standard product line available "off the shelf." They have been designed for such applications as on-site power for remote locations or for temporary installations.

Depending on the amount of waste energy available for conversion into electrical power at a given time of day, one or more generators such as those described in the preceding paragraph could be switched "on-line" to take up a part of the electrical load otherwise drawn from commercial sources. At the power and voltage levels available in such a system (2.5 megawatts/4160 volts), a sophisticated synchronizing switching system with full personnel and equipment protection features normally would be required. However, an alternative is available and is recommended for use in the jet engine test cell waste energy recovery system. The alternative is semi-automatic switching and offers full personnel and equipment safety features.

Relay supervision of manual switching, or semi-automatic switching, allows personnel with a minimum of training to safely bring a generator system into full parallel operation with the main power lines, provide motoring or reverse current protection to the generator and automatically monitor current and voltage output levels and provide the circuit breaker function for the system. Relay supervision of manual switching is recommended over a fully manual switching system because of the training required of personnel to safely bring the test cell generator into phase with the primary system. In addition, a semi-automatic switching requires a far smaller capital investment than a fully automatic system.

A brief summary of the recommended electrical generator and switching system specifications are:

TYPE ATI Drip-proof Generator

WITH: brushless exciter
static voltage regulator

3125 KVA

2500 KW

4160 V, 3Ø, 60 HZ

1800 RPM

Full load efficiency = 95.9%

Operates sea-level to 3300 feet

Estimated price (including auxillary equipment)	$70,800
POWER/VAC Semi-automatic switchgear 4160 V, 3Ø, 60 HZ	
Estimated price	45,000
Total Estimated Price	$115,800

CONCLUDING SUMMARY AND RECOMMENDATIONS

The base surveys have indicated that much can be done with regard to waste energy recovery. In nearly every example given in this report "off-the-shelf" equipment that can be selected from any of several manufacturers, is adequate to accomplish the task. In most of these examples, even for rapid break-even times (i.e., less than five years), the annual energy savings are small amounts. If such recovery schemes, however, were implemented throughout the Department of Defense, annual savings would surely reach into the tens of millions of dollars. As energy costs escalate or energy cutbacks occur, this energy recovery equipment would then become more economically and politically favorable. In addition, environmental pollution is reduced because of the reduction in energy consumption.

The study of the Ram Air Test Facility at Tinker AFB (c.f. Section 12) is an excellent example of the application the methodology presented in Section 3. Quantification of the energy use and construction of the energy flow diagram provided immediate indication of the potential for energy recovery. By considering the facility as a system, the multiple requirement to cool compressed air and subsequently reheat it lead to the proposed use of a gas turbine unit. The gas turbine unit not only provides the shaft work for the air compressor, but the exhaust gases are capable of reheating the compressed air. By considering Tinker AFB as the system, it was discovered that waste jet fuel could be used as the fuel

for the gas turbine, further reducing costs and energy consumption. This kind of layered systems approach should be used on all projects to insure the optimum use of all potential sources and the coupling of these sources to the best locations. The gas turbine unit and the filtration system alone is projected to result in accumulated savings in utility costs of $1.7 million over the next ten years for a total initial capital outlay of $340,000 and an average annual maintenance cost of $6,000.

One of the important finds of the study was the identification of three critical factors which are necessary for the success of energy recovery projects. The first of these is the annual fraction of time the rejected energy source will be utilized must be large. Secondly, the physical distance that the energy must be transported must be short. Lastly, the required maintenance for the recovery equipment must be small compared to the initial annual energy savings. The studies have indicated that if any of these three factors were not satisfied, then the proposed use of the rejected energy will not be economically practical. At the Ram Air Test Facility, for example, use of the energy currently dumped to resistor banks during air turbine testing for power generation, battery charging, and/or heating, was contemplated. Although transmission of the source and maintenance were not critical, the intermittent nature of the source resulted in excessively long payback periods. This was true even in the case of resistance heaters for compressed air heating.

The design of new equipment should be avoided, because, current costs will outweigh any potential energy savings. Most heat recovery equipment available today has either been in existence for some time or is merely a modified size or shape of proven units normally used for other purposes. The heat-pipe exchanger has been known to be an excellent heat exchanger, but until recently, it has not been economically viable. It is recommended, because of their high efficiency and low maintenance, that heat-pipe exchangers be considered in energy recovery studies. They have been shown to be especially useful for air-to-air heat recovery.

The difference between energy quantity and quality is an extremely important concept. While energy quantity indicates the amount of BTU's, KW, etc., rejected energy quality indicates the degree of usefulness of the energy. In the full thermodynamic sense, energy quality is measured by the availability function which is a measure of a system's ability to do useful work. Since energy recovery through rejected energy usually involves heat transfer, however, temperature was found to be a good indicator of energy quality. High quality sources are favored, because they lead to compact, often low-cost, heat exchangers, and hence, rapid payback periods. Oftentimes, however, the rejection of high quality energy is the result of an inefficiency, which, when corrected, reduces the quality and often the quantity of the rejected energy. As a result, the majority of rejected energy sources will be low quality sources. Effective recovery of the low quality sources demands careful analysis, since one deals in small differences of relatively large numbers.

Most Air Force facilities lack the proper energy monitoring equipment to determine the energy consumption of individual facilities. This is an understandable omission however, because most Air Force facilities were constructed when energy was inexpensive and thought to be inexhaustible. Energy monitoring equipment can: (a) help identify large energy users, (b) provide the necessary data to quantify the justification of the installation of energy recovery equipment, and (c) monitor the relative success or failure of the energy recovery equipment. Blanket installation of energy monitoring equipment at any Air Force Base is not the sensible approach, because the initial capital costs would be excessive. Major energy users, however, are usually known, and they should be considered first. It is further recommended that energy monitoring equipment be installed before energy recovery equipment, so that relative savings can be charted.

The study and implementation of energy recovery equipment would be the responsibility of the Base Civil Engineer working in conjunction with the energy user. Current duties of these personnel usually limit their ability to rapidly implement such a program. They are not only hampered by their daily duties, but they will be required to spend an appreciable amount of time learning about waste energy recovery before they could implement the program. It seems appropriate, therefore, that a program be established to help base personnel accomplish an effective energy recovery program. Such a program could be established and controlled by one central agency or several regional offices. It is important at this time that a central agency monitor and control the efforts because of the changing developments in recovery equipment and its use. It is also suggested that the central point or regional points

maintain a strong liaison with their appropriate counterparts in the Navy and Army to reduce potential duplication. Furthermore, a liaison with the appropriate section in the Office of the Assistant Secretary for Conservation and Solar Applications of the Department of Energy should be maintained. This could be accomplished through annual or semi-annual meetings.

The focal point(s) should use the services of architect-engineer (A&E) firms to conduct waste energy surveys and design the most promising projects identified. These A&E firms, working with the Base Civil Engineer will conduct surveys, prescribe energy monitoring and energy recovery equipment, and determine economic feasibility of the projects. Surveys should be complete enough so that the Base Civil Engineer would merely have to contract for the services suggested, if funding is approved. Such a program would insure uniformity of effort, minimum expense for surveys, and avoid duplication. Effort of the A&E firms could be further minimized if the examples in this report were made available to them, and they documented additional examples resulting from their studies.

Secion 14 of this report was devoted to some of the problems and potential energy recovery schemes for the jet engine test cells at Tinker AFB. The amount of energy consumed is significant, and they alone could have an appreciable effect on the bases' energy consumption. Currently, jet engine test cell facilities throughout the country are merely big "mufflers" that suppress jet engine noise and insure proper cooling of the tailpipe. There is not only a significant potential for energy recovery at these facilities, but such equipment could be combined with the appropriate pollution control devices. Both of these factors would justify the installation of virtually any type of energy recovery and/or pollution control device.

REFERENCES

1. Lichtenburg, S.J., "Energy Efficiency and Energy Savings", Energy, Vol. 2, pp. 283-286.

2. Reynolds, W.C. and H.C. Perkins, Engineering Thermodynamics, Mcgraw-Hill Book Company, New York, New York, 1970.

3. Van Wylen, G.J. and R.E. Sonntag, Fundamentals of Classical Thermodynamics, 2nd Edition, John Wiley and Sons, New York, New York, 1973.

4. Berg, C.A., "A Technical Basis for Energy Conservation", Technology Review, February 1974, pp. 15-23.

5. Kent's Mechanical Engineer's Handbook, Power Volume, Twelfth Edition, Wiley Engineering Handbook Series, John Wiley and Sons, New York, New York, 1950, pp. 7-36.

6. Adams, P., "Air Force Commissary Energy Management Specification Guideline", AAFES-CD-S, HQ 77-CDS-018, April 25, 1977.

7. Beenhakker, H.L., "Sensitivity Analysis of the Present Value of A Project", The Engineering Economist, Vol. 20, No. 2, 1975, pp. 123-150.

8. Hise, E.C. and A.S. Holman, "Raising the Efficiency of Gas Furnaces", Oak Ridge National Laboratory Report, January 1976.

9. American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc, Handbook of Fundamentals, Fourth Printing, New York, New York, 1972, Chapter 20.

10. Energy Alternate Source Team, Deputy for Plans and Programs Report, Tinker AFB, OK.

11. Pai, S.I., Fluid Dynamics of Jets, D.van Nostrand Company, New York, New York, 1954

APPENDIX A

Development of the Necessary Derivatives for Determining the Uncertainties in the Break-Even Economics

The present

The break-even time, n, is a function of five independent variables, initial capital cost, C, annual maintenance costs, M, initial annual energy savings, G_o, utility escalation rate, i_u, rate-of-return, i^*. The variables are functionally related by the following equations:

$$C + n\,M - \frac{G_o\,(1 + I)\,\{(1 + I)^n - 1\}}{I} = 0 \qquad \text{(A-1)}$$

where,

$$I = \frac{1 + i_u}{1 + i^*} - 1 \qquad \text{(A-2)}$$

It is desired to obtain the derivatives of the break-even time with respect to the five independent variables. Since the break-even time cannot be solved for explicitly, the derivatives of interest must be determined through implicit differentiation.

The derivatives with respect to C is as follows:

$$\frac{\partial}{\partial C}\left[C + n\,M - \frac{G_o\,(1 + I)\,\{(1 + I)^n - 1\}}{I} = 0\right] \qquad \text{(A-3)}$$

$$1 + M\,\frac{\partial n}{\partial C} - \frac{G_o\,(1 + I)^{n+1}}{I} \cdot \ln\,(1 + I)\,\frac{\partial n}{\partial C} = 0 \qquad \text{(A-4)}$$

Solve for $\frac{\partial n}{\partial C}$,

$$\frac{\partial n}{\partial C} = \frac{1}{G_o F_1 - M} \tag{A-5}$$

where,

$$F_1 = \frac{(1 + I)^{n+1}}{I} \ln (1 + I) \tag{A-6}$$

Similarly for M,

$$\frac{\partial n}{\partial M} = \frac{n}{G_o F_1 - M} \tag{A-7}$$

for G_o,

$$\frac{\partial n}{\partial G_o} = - \frac{(1 + I) \{(1 + I)^n - 1\}}{I (G_o F_1 - M)} \tag{A-8}$$

and for G_o',

$$\frac{\partial n}{\partial I} = + G_o \frac{(1 + I)^n (nI - 1) + 1}{I^2 (G_o F_1 - M)} \tag{A-9}$$

The derivatives of n with respect to i_n and i^* require the differentiation of equation (Δ - 2):

$$\frac{\partial I}{\partial i_u} = \frac{i}{1 + i^*} \tag{A-10}$$

$$\frac{\partial I}{\partial i^*} = - \frac{1 + i_u}{(1 + i^*)^2} = - \frac{I}{1 + i^*} \tag{A-11}$$

Therefore,

$$\frac{\partial n}{\partial i_u} = \frac{1}{1 + i^*} \frac{\partial n}{\partial I} \tag{A-12}$$

and

$$\frac{\partial n}{\partial i^*} = - \frac{I}{1 + i^*} \frac{\partial n}{\partial I}$$

APPENDIX B

B I B L I O G R A P H Y

The following list of books, government reports, articles and archive journal publications has been compiled as a result of this study. Although this list is not a complete listing of information germane to energy recovery, it should provide the uninformed reader with substantial information on the subject.

A. Texts

1. American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., New York, New York, 1972, ASHRAE Handbook of Fundamentals, Fourth Printing.

2. Kays, W.A. and London, A.L., Compact Heat Exchangers, 2nd Edition, McGraw-Hill Book Company, New York, New York, 1964.

3. Welty, James R., Engineering Heat Transfer, John Wiley and Sons, New York, New York, 1974.

4. Reynolds, William C. and Perkins, Henry C., Engineering Thermodynamics, McGraw-Hill Book Company, New York, New York, 1970.

5. Wiley Engineering Handbook Series, Kent's Mechanical Engineer's Handbook, Power Volume, Twelfth Edition, John Wiley and Sons, New York, New York, 1950.

6. Boyen, John L., Practical Heat Recovery, John Wiley and Sons, New York, New York 1975.

7. Kreith, F., Principles of Heat Transfer, 2nd Edition, International Textbook Company, Scranton, Pa.

8. Giedt, Warren, Thermodynamics, Van Nostrand Reinhold Company, New York, New York, 1971.

B. Handbooks and Pamphlets

9. An Approach for Managing an Energy Conservation Program, April 1977, Air Force Civil Engineering Center, Tyndall AFB, Florida.

10. Basic Information on the Economic Generation of Energy in Commercial Quantities from Wind, Engineering Energy Laboratory, Oklahoma State University, May 1974.

11. Building Energy Handbook, Volume 1, ERDA-76/163/1, December 1976, Energy Research and Development Administration, Washington, D.C.

12. Energy Conservation Program Guide for Industry and Commerce, NBS Handbook 115, Federal Energy Administration, Washington, D.C.

13. Energy Management Services, Air Force Commissary Energy Management Specification Guideline, April 1977.

14. Heat Pipes, AIAA Selected Reprint Series, Vol. 16, American Institute of Aeronautics and Astronautics, New York, New York.

15. Identifying Retrofit Projects for Federal Buildings, Federal Energy Management Program, Federal Energy Administration, Washington, D.C.

16. Lighting and Thermal Operations, Federal Energy Administration, Washington, D.C.

17. Optimization of Energy Usage in Military Facilities (Phase 1), Air Force Civil Engineering Center, Tyndall AFB, Florida.

18. Policies, Procedures and Criteria for the Management and Conservation of Utilities, AFM 91-12, August 1971, Department of the Air Force.

19. Research Needs Report: Energy Conversion Research, American Society of Mechanical Engineers, New York, New York.

20. Retrofitting Existing Housing for Energy Conservation: An Economic Analysis, Building Science Series 64, Federal Energy Administration, Washington, D.C.

21. Technical Guidelines for Energy Conservation, Air Force Civil Engineering Center, Tyndall AFB, Florida, June 1977.

22. Technical Guidelines for Energy Conservation in New Buildings, January 1975, Naval Facilities Engineering Command, Alexandria, Virginia.

23. Waste Heat Management Guidebook, NBS Handbook 121, Federal Energy Administration, Washington, D.C.

C. Publications

24. Bachtel, W.D., "Heat Recovery from Plant Process Exhaust", Heating/Piping/Air Conditioning, V. 43, May 1971, pp. 88-92.

25. Bjerklie, J.W., "Small Gas Turbines for Total Energy Systems", Actual Specifying Engineer, V. 26, August 1971, pp. 638-641.

26. Black, T.W., "Advanced Turbine Designs Boost Wind-Power Potential", Machine Design, June 1976, pp. 26-31.

27. Bridgers, F.H., "Energy Conservation: Pay Now, Save Later", ASHRAE Journal, V. 15, October 1973. p. 47.

28. Brown, S.W. and E. Gore, "Energy Conservation by Low Temperature Waste Heat Recovery", Building Systems Design, V. 69, June 1972, pp. 4-7.

29. Burger, Robert, "Cooling Tower Rebuilding", Cooling Tower Institute Annual Meeting, January 1975, New Orleans, LA.

30. Calm, James M. and William J. Coad, "COP vs. Energy Efficiency", Heating/Piping/Air Conditioning, August 1977, pp. 70-75.

31. Cantrell, C.J., "Waste Energy Conservation Techniques", ASMA publication 76-PET-21, American Society of Mechanical Engineers, New York, New York.

32. Cooper, G.P., "Energy Conversion Research - An Assessment cf Need in The U.S.A.", ASME publication 76-WA/RFPC-6, American Society of Mechanical Engineers, New York, New York.

33. Cremers, C.J., "Environmental and Conservation Research Needs in the Eighties", ASME publication 76-WA/RGPC-2, American Society of Mechanical Engineers, New York, New York.

34. Daryanani, S., "HVAC Designs for Saving Energy", ASHRAE Journal, V. 15, February 1973, pp. 32-33.

35. Davis, W.J., "Heating of Industrial Buildings", ASHRAE Journal, V. 15, November 1973, pp. 40-44.

36. Doherty, M.C., "Investment Economics of Industrial Gas Turbines", ASME publication 77-GT-26, American Society of Mechanical Engineers, New York, New York.

37. Fanaritis, J.P. and H.J. Streich, "Heat Recovery in Process Plants", Chemical Engineering, V. 80, May 1973, pp. 80-88.

38. Fergin, R.K., "Heat Recovery Devices for Air Conditioning", Heating/Piping/Air Conditioning, V. 44, April 1974, pp. 100-102.

39. Gourdine, M.C. and D.H. Malcom, "Prospect for Electrogasdynamic Power", Gourdine Systems, Inc., Livingston, NJ.

40. Green, Leon, Jr., "Gas Turbine Total Energy System for A Cement Plant", ASME publication 77-GT-22, American Society of Mechanical Engineers, New York, New York.

41. Green, Robert W., "Heat Recovery Water Heating", Patrick AFB, FL.

42. Lichtenburg, Allan J., "Energy Efficiency and Energy Savings", Energy, Vol. 2, pp. 283-286.

43. Mills, L., "Recover Waste Heat Systematically", Power, V. 116, December 1972, pp. 36-37.

44. Pannkoke, Ted, "Packaged Equipment for Air-to-Air Energy Recovery", Heating/Piping/Air Conditioning, August 1977, pp. 43-45.

45. Polimeros, G., "Some Considerations in Industrial Heat Recovery", Building Systems Design, V. 70, April 1973, pp. 36-38.

46. Reynolds, B.L. and R. E. Holmes, "Heat Transfer in A Corona Discharge", Mechanical Engineering, October 1976, pp. 44-49.

47. Ritche, D.A., "Water Heating by Refrigeration", Weatherking, Inc. Orlando, FL.

48. Robertson, J., "Energy Conservation in Existing Plants", Chemical Engineering, V. 81, Jan. 1974, pp. 104-111.

49. Tassicker, O. and J. Schwab, "High Intensity Ionizer for Improved ESP Performance", EPRI Journal, June/July 1977, pp. 56-61.

50. Trezek, G.J. and D.M. France, "One-Dimensional Particulate Electrogasdynamics", AIAA Journal, Vol. 8, No. 8, pp. 1386-1391.

51. Wallinger, L., "A Comparison of Steam Vs. Freon Cycles in Waste Heat Recovery from Stack Gases", Sun Power Systems, Miami, FL.

52. Walters, Ted, "Energy Recovery Solves Hospital's Problem", Heating/Piping/Air Conditioning, August 1977, pp. 54-56.

53. "Heat Recovery Water Heating Systems", Application Engineering and Research Section, Florida Power Corporation, May 1972.

54. "Waste Heat - and How It Might Be Used", Aware, V. 27, December 1972, p. 11.